

# THE EFFICIENT USE OF FUEL

## ERRATA

- Page 24, lines 13 & 15. For "carbon per cent." read "100."
- Page 46, line 31. For "experience" read "laboratory Experiment."  
(This correction has been made in some copies of the Handbook.)
- Page 53. 4th line from bottom should read "Total waste gases (wet) =  $138.81 + 65.7 = 204.51$ ."
- Page 54, table 15. 6th line of figures should read :  
11.65 ; 154 ; 160 ; 17.5 ; 76.5 ; 6.0 ; 18.6.  
Ditto. 7th line of figures, for "74.1" read "74.5."
- Page 62, 5th line under table. For "44.31 B.Th.U." read "44,310 B.Th. U."  
(This correction has been made in some copies of the Handbook.)
- Page 80, Fig. 21. 5th number up the ordinator, for "0.8" read "0.6."
- Page 81, 11th line. For 2°F. read 1°F.
- Page 122, last 2 lines. For "1240," "1050," and "1145" respectively, read "1730," "2000" and "1860."
- Page 125, lines 6 and 7. For units in which  $h_v$  and  $h_w$  are expressed, viz.  
"B.Th.U./sq. ft./hr. °F./ft." read "B.Th.U./sq. ft./hr. °F."
- Page 127, line 11. Read "If the *outside* diameter of a pipe . . . etc."  
Line following eq. (7) should read : "where  $t_1$  and  $t_2$  are the temperatures at diameters  $d_1$  and  $d_2$  respectively, and  $k$  is expressed as B.Th.U./sq. ft./hr. °F./ft. thickness."  
Line following eq. (8) should read : "where  $r$  is the distance from the pipe axis in feet when  $k$  is expressed in units of 1 foot thickness ;  $r$  is in inches when  $k$  is in B.Th.U. per inch thickness, when  $r = \frac{1}{2}d_1$  . . . etc."  
Next aparagraph and worked example :  
for "105°F." read "120°F."  
for "0.42" read "0.035."  
for "1.725" read "140."  
for "1096" read "89."
- Page 337. Footnote under table 68 should read : "(The above figures refer primarily to water-tube boilers ; for other boilers working below 250 lb./sq. in., the maximum T.D.S. recommended is : Economic boilers 400 parts/100,000 ; Lancashire and Cornish, 1500 parts/100,000)."
- Page 343, line 3. For "sensible heat in flue gases" read "Loss through sessible heat in flue gases, per cent."
- Page 519, line 6. For "Table 101" read "Table 100."  
(This correction has been made in some copies of the Handbook.)
- Page 547, line 19. For "If a 5-inch steam pipe . . . etc.", read : "If a 1-inch steam pipe, . . . etc."  
line 22. For "70 tons" read "105 tons."
- Page 680, Fig. 259. The first and third jets should be transposed. The latter inverted.
- Page 805. Index. For "Pilot" read "Pitot."

MINISTRY OF FUEL AND POWER  
July, 1945.

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# THE EFFICIENT USE OF FUEL

A TEXT-BOOK ON FUELS AND THEIR EFFICIENT  
UTILISATION FOR THE USE OF STUDENTS AND  
TECHNICAL MEN IN INDUSTRY

Prepared under the Direction of the Education  
Sub-Committee of the Fuel Efficiency Committee  
of the Ministry of Fuel and Power

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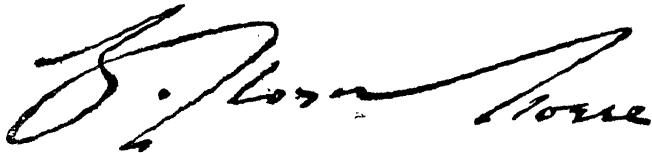
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## FOREWORD

FROM THE MINISTER OF FUEL AND POWER

THE effective use of fuels of every kind is of vital importance now, and will continue to be essential to the country's well-being so long as we are dependent on coal for our heat and power. The promotion of the efficient use of fuel has been a major preoccupation of the Ministry of Fuel and Power since its formation, and I trust that this book, in which has been assembled a very great deal of information not readily available, will not only serve the present urgent need, but will be of permanent value to all those concerned with the industrial use of fuel.

A handwritten signature in black ink, appearing to read 'B. Horne'.

WESTMINSTER HOUSE,  
7 MILLBANK, S.W.1.  
*July,*

## PREFACE

MANY and hard are the lessons taught by the stress and strain of war. Not the least amongst those that emerge from the present conflict has been the realisation by the British people of the value of their coal resources. There is a danger that the supply of fuel will be inadequate to maintain industrial activity to its fullest extent, and, as a consequence, the vital supply of munitions will be interrupted. To meet this situation the Fuel Efficiency Campaign has been organised by the Ministry of Fuel and Power; and amongst the instruments of that movement were the emergency training schemes undertaken jointly by the Board of Education and the Ministry to spread the knowledge of the efficient methods of using fuel.

It was soon realised that no single existing text-book covered both the fundamental principles and the essential operative features necessary to attain immediate results. The help of leading fuel technologists was therefore enlisted by the Education Sub-Committee of the Fuel Efficiency Committee, and a syllabus for the guidance of lecturers and a series of lecture notes that could be given to students were drawn up. In the course of preparing this material, however, it was soon realised that only the barest information could be given and the urgent need for a more comprehensive text-book on Fuel Efficiency was disclosed. Moreover, the favourable reception accorded to the Sub-Committee's earlier effort was a further incentive to the preparation of an extended and illustrated text-book, embracing the use of all fuels for industrial purposes.

The present work is the outcome of this effort. Its preparation has been entrusted by the Sub-Committee to the General Editorial supervision of Dr. G. E. Foxwell. Many leading authorities have generously assisted in its compilation and Dr. Foxwell has himself contributed substantially to the preparation of the subject matter. Each individual chapter has been submitted for approval to panels of experts on the particular subjects under review, and they have in many cases subjected them to certain criticism and revision. It is thus apparent that every means has been taken to ensure that the book is as authoritative as possible, free from personal bias and appreciative of the practical outlook.

It is not intended as a book of reference to be placed on the shelves of libraries and to which occasional reference is made. Each chapter is meant to be read by the student, and though it may be that one individual will not read them all, he will select those in which he is interested, and possibly become interested in some of the others.

The book was originally intended to emphasise the "application of knowledge," and arising out of this attempt to be practical, the preparation of certain chapters has indicated where information is lacking, and the Committee hopes to explore these untrodden paths in the near future.

In spite of the efforts made to present an effective work, the book has been prepared under emergency conditions and the need for haste has been uppermost. It has been done by busy men and many imperfections may accordingly be apparent. For these indulgence is asked, and it is requested that they may be pointed out so that they may be removed or corrected in future editions.

Further, it is hoped that this work may be regarded as part of a live and progressive movement, directed not only to the betterment of the present

supply situation, but to an improvement in the efficient use of our greatest mineral asset. It may not be possible to say how long our coal resources will last, but it is possible to ensure that the most may be made of their potential value.

The Education Sub-Committee cannot refer too highly to the skill and enthusiasm shown by Dr. Foxwell in the editing of this work. Their cordial thanks are offered to Mrs. Fuller for her share in piloting the work through its preliminary drafts, revision and proof reading.

Grateful acknowledgement is accorded to those mentioned below, for having taken part in the production of this work, and for their generous and able assistance, always given so freely and promptly. Thanks are extended, in no less a degree, to all those who have assisted but remain anonymous.

*R. J. Sarjant*

*E. S. Grumell*

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## CHAPTER I

# INDUSTRIAL FUELS : COAL

Fuel conservation—Origin and nature of coal—Properties of coals—Grouping of coals—Main uses—Industrial characteristics—Sulphur—Coal preparation and cleaning—The size of coals.

WITH the exception of petroleum oils, substantially all fuel used in this country either consists of raw coal or is manufactured from coal ; coal is therefore the basis of fuel and power in Great Britain.

## FUEL CONSERVATION

Fuel conservation has two main aspects ; (1) As a long-term policy affecting a national asset of irreplaceable raw material. (2) As a short-term policy to meet the urgent need for reducing consumption during the war.

The importance of the long-term policy to posterity requires a little attention. According to Bone and Himus,\* the net coal which will be available at the surface will be about 170,000 million tons. This estimate is that of Stanley Jevons, in 1915, of reserves within 4,000 feet of the surface, taking 1 foot as the minimum workable thickness of seam.

In 1913, 287 million tons of coal were mined, and in 1938, 227 million tons. It might therefore be supposed that coal will last from 580 to 749 years at these rates of production. This gives an agreeable sense of security but, as Bone and Himus point out, " the most important aspect of the coal question for Great Britain is not so much how long can our reserves last, but how long can we continue to get the available coal at a cost which shall not place us as a nation at a disadvantage relative to our nearest competitors." The answer is not something of the order of 500 to 800 years, but possibly some 50 years or less. A more recent analysis of the position in 1940 by the then President of the Institution of Mining Engineers, Mr. Forster Brown (Presidential Address, 1940) fully confirms this estimate of the position and emphasises the probable increase in coal-getting costs. A shortage of certain special types of coal, such as coking coals, may well be experienced within the next two generations. Early consideration will have to be given either to forbidding the use of these coals, except for special purposes for which they are peculiarly adapted, or to finding a means of making other types of coal suitable for coke manufacture.

From the war aspect—the short-term policy—the maximum effort must be made now to use fuel with the greatest efficiency. The object of this handbook is to assist in attaining this goal ; at the same time the necessity for reorganisation of the methods of coal preparation and utilisation after the war is clear.

## THE ORIGIN AND NATURE OF COAL

The term " coal " is applied to those rocks in the earth's crust produced by the decay of plant materials, accumulated millions of years ago. Shales impregnated with carbonaceous substances are excluded. Scientists have identified in coal, traces of the structure of known plants and even portions of the trees themselves, which have suffered only a slight change of form.†

\* " Coal, Its Constitution and Uses," p. 17, Longman, Green & Co., London, 1936.

† J. G. King: " The Preparation, Selection and Distribution of Coal," Chapter I (The Coal Trade Luncheon Club and the Institute of Fuel, 1931).



Two theories have been suggested to explain the manner in which these accumulations of plant remains have collected in sufficient quantity to form coal seams. The first—the growth in place or “in situ” theory—holds that the coal seam marks the area in which the original material grew and accumulated. The second—the “drift” theory—holds that the material was transferred by rivers from the area in which it grew, to a lake or estuary in which it was deposited. It is probable that both methods of accumulation have been operative.

The best example of a thickly forested fresh-water swamp, illustrating the “in situ” theory, is the Great Dismal Swamp of Virginia, North Carolina (U.S.A.) The water is shallow and the soil below consists of 10–15 feet of partially decayed trunks, branches, leaves, roots and seeds of the continually falling trees; sand and other impurities are absent. Coastal swamps of a similar nature are abundant in the tropics.

As an example of the “drift” method may be cited the large quantities of material consisting of trunks, branches, etc. carried by the Mississippi—particularly at flood time—to the mouth of the river where the water-logged condition of such material causes it to be deposited. As it may happen that at the mouth of the river there is already growing an estuarine swamp, it will be seen that both methods may contribute to the same area of accumulation.

On dry ground, fallen trees and other dead plants are attacked by the oxygen of the air, and the cellulose, their chief constituent, is converted into water and various gases so that in a comparatively short time they have rotted away, leaving practically no trace. Under water, however, when air is excluded, plant debris is decomposed by the action of bacteria. This decomposition continues until the substances produced become so concentrated that the bacteria can no longer live; the extent of the decomposition, therefore, depends on whether or not the products can drain away. Certain parts of the plant are always more resistant to decay than others, and fragments of plants only slightly altered, particularly spore exines (the tough outer jackets of spores), bark, or its cuticle (outside skin) can be identified in coal.

As a result of geological agencies which caused the surface of the earth to fall and rise alternately, the accumulations of partly decayed plant material have been covered by layers of sedimentary rocks, and then further plant remains have been deposited. In this way a sequence of seams has been built up.

The changes that took place after the cessation of bacterial decay may have been even more profound. The influences then operating were heat and pressure, the pressure being exerted not only vertically by the weight of the overlying strata, but also sideways by the powerful earth movements which have resulted in the folding and compression of the seams. The chemical action which took place was the elimination of water and oxides of carbon, resulting in a reduction of the amount of material that can be obtained from the coal in the form of volatile vapours and gases (the “volatile matter”) by the action of heat. There is some doubt how far these subsequent changes in the coal seams have been due to heat. Normally, heat would be indirectly connected with the pressure, but the temperature of parts of a few seams has been raised by molten igneous intrusions from below.

## VARIATIONS IN COAL

The result of the action of natural agencies upon the plant remains has not always been the same. These variations may be considered under the following headings:—

- (1) The nature of the original vegetation and its separation by the grading action of water.
- (2) The extent of bacterial decay.
- (3) The extent of the removal of volatile matter by heat and pressure.

## INDUSTRIAL FUELS : COAL

(1) Continuous seams of coal are known, for example in South Wales, in which the character of the coal changes steadily in a lateral direction, from bituminous to anthracite (see later) ; microscopical examination shows that the same types of plant were present over the whole area. The difference cannot, therefore, be accounted for by changes in vegetation. The segregation of different parts of similar vegetation can, however, produce a marked difference in composition. For example, the dull bands (durain) which occur in many seams contain more plant remains than the bright bands adjacent to them. Other plant fragments also bear evidence of having been water borne. This suggests that some segregation of the parts of the plant by the action of water occurred and produced a difference in the final coal.

(2) The extent to which bacterial action proceeded was probably influenced by drainage, since the decomposition normally continues until the products accumulate to such an extent as to arrest decay.

(3) The heat and pressure, to which a coal has been subjected would depend upon the depth of the superimposed strata and the degree of earth movement. In areas where the seams have remained relatively undisturbed, the oldest are frequently lowest in volatile matter, the percentage tending to decrease with depth. This was stated by Hilt to be of fairly general application, but several noted exceptions occur. In the South Wales seams mentioned above, the anthracitisation occurs in the north-west of the coal-field where there has been most disturbance. It is, therefore, possible that pressure and heat are responsible for the change. In Scotland, parts of certain seams are changed to a pseudo-anthracite by an entirely different cause, viz., by the heat derived from the intrusion of molten magma.

## GLOSSARY OF TERMS—SOME CONSTITUENTS AND CHARACTERISTICS OF COALS

Before proceeding to discuss the types of coal available it is necessary to explain certain terms which are used when describing coals.

*Moisture.* The amount of moisture retained by a coal may vary between fairly wide limits. By exposing coal in a thin layer in a dry, well-ventilated place, the coal loses " free " or " surface " moisture, and dries to an " inherent " or air-dried moisture content in equilibrium with the moisture in the atmosphere. At a given humidity, the inherent moisture content of coal, determined by heating the air-dried coal to 110° C., is characteristic of that particular type of coal and varies, with different British coals, from below 1 per cent. to about 15 per cent. The free or surface moisture is mechanically adhering moisture which may come from the operation of wet cleaning processes or which may be due to exposure to rain.

*Ash.* The ash in industrial coal is determined by the complete combustion of a weighed sample. It includes the " inherent " ash intimately associated with the coal and partly or mainly derived from the original coal plants, and adventitious ash derived from shale, clay, pyrites, ankerites (white partings) or dirt from bands in the coal seam, and fragments of the roof or floor which have become included in mining the coal. The adventitious ash-forming constituents can be removed by physical means, but the inherent ash-forming constituents cannot be removed in this way. The ash in British coals cleaned by floating in a solution of specific gravity of 1.6 amounts, on the average, to between 4 and 4.5 per cent., ranging from 1 to 8 per cent. for individual coals. The ash in British coals cleaned by floating in a solution of specific gravity of 1.35 amounts, on the average, to about 2.8 per cent., ranging from 1 per cent. to about 5 per cent. for individual coals.

The amount of ash as determined is less than the mineral matter originally present by the amount of water and carbon dioxide driven off from clays

during combustion, and due to other minor changes, as, for example, the oxidation of pyrites. Generally it can be said that the weight of ash is about 10 per cent. less than the weight of original mineral matter, i.e. if the amount of "ash as determined" is 20 per cent. the amount of inert mineral matter frequently referred to as "ash" is really 22 per cent.

The amount of ash and its fusion temperature have an important bearing upon the value of coal in industry (see Chapter VI).

*Volatile Matter.* The volatile matter which, it should be made clear, does not include the moisture in the coal, is defined as the percentage loss in weight when 1 gm. of coal is heated under carefully controlled conditions for seven minutes at 925° C. in a crucible from which air is excluded (cf. Chapter XXXIII).

The loss consists of water (derived from the decomposition of the coal and not from surface, or inherent, moisture), gas and tar; the percentage of volatile matter (on the dry coal) when expressed on the dry ash-free basis, supplies considerable information as to the probable behaviour of the coal in use.

*Dry Ash-Free or Dry Mineral-Matter-Free Coal.* For the purpose of comparing certain characteristics of coal, and for grouping coals, it is customary to use figures based on dry ash-free coal, the comparison then being on the actual coal substance irrespective of adventitious material; for instance, if a coal contains 5 per cent. moisture and 5 per cent. ash—a total of 10 per cent. of material which can be termed "inert"—and has a calorific value of 12,000 B.Th.U., this value on the dry ash-free basis will be :—

$$12,000 \times \frac{100}{90} = 13,333 \text{ B.Th.U.}$$

Similarly, if the determined amount of volatile matter (less moisture) be 36 per cent., it would be  $\frac{36}{0.9} = 40$  per cent. on the dry ash-free basis. To represent approximately the mineral matter, the weight of ash should be increased by 10 per cent., and the calculation will then give results on a dry mineral-matter-free basis.

For exact scientific work the dry mineral-matter-free basis should be used. One formula that can be used for this purpose is due to King, Maries and Crossley, which states :—

$$\begin{aligned} \text{Mineral matter} = & 1.09 \times \text{per cent. ash} + 0.5 \times \text{per cent. pyritic sulphur} + \\ & 0.8 \times \text{per cent. CO}_2 \text{ evolved on treatment with acid} - \\ & 1.1 \times \text{per cent. SO}_3 \text{ in ash} + \text{per cent. SO}_3 \text{ in coal} + \\ & 0.5 \times \text{per cent. chlorine in coal.} \end{aligned}$$

A simpler method is due to Parr :—

$$\text{Mineral matter} = 1.08 \times \text{per cent. ash} + \text{per cent. moisture} + 0.55 \times \text{per cent. sulphur.}$$

The King-Maries-Crossley formula assumes the coal to be dry. The Parr formula takes the moisture into account and also assumes that all sulphur is present in the form of pyrites.

*Fixed Carbon.* The residue left after heating the coal in a closed vessel to drive off the volatile matter is the "coke." This residue contains all the inorganic constituents present in the original coal that go to form the ash.

The percentage of solid residue minus the ash = "fixed carbon."

The solid residue also contains hydrogen, sulphur, nitrogen and oxygen in addition to carbon. It may vary from a black powder to a highly porous button, and its appearance, strength and cellular structure are significant characteristics.

*Caking and Swelling Power.* The appearance of the residue from the volatile matter determination is a guide to the property which a coal possesses of

## INDUSTRIAL FUELS : COAL

forming a coherent coke. A more reliable method of assessing this property is to use an assay test, one example of which is the Gray-King.\* In this assay, 20 gm. of the coal are heated in a silica tube in a special furnace to 600° C. (1,112° F.). The appearance of the resulting residue (shown in Fig. 1) is then compared with a series of standard cokes and its position determined in the series, which ranges from non-caking up to highly-caking coals. The letters A to G are used to designate cokes which range from those given by a non-caking coal (Type A) to those which are hard and strong and are of the same volume as the original coal (Type G). For coals more strongly swelling than G, subscripts are added, e.g. G<sub>1</sub>, G<sub>2</sub>, G<sub>3</sub>, etc. up to G<sub>10</sub>, in order to indicate the degree of swelling.

The swelling properties may be determined by the Woodall-Duckham crucible swelling test described in British Standard Specification 1016 of 1942, page 64. The remarkable differences in the swelling power of coals are illustrated in Figs. 2 and 3 of coke buttons obtained in the Woodall-Duckham swelling test and in the Lessing method of determination of volatile matter.

The relationship between the results of these methods is as follows :—

W-D crucible swelling test	Gray-King	Lessing
1	A-B	1
1½	C-D	1½
2 - 2½	D-E	2
3	D-G	2½
4 - 4½	G <sub>1</sub>	3
5 - 5½	G <sub>2</sub>	3½
6½ - 7	G <sub>3</sub>	4
9	G <sub>4</sub> -G <sub>10</sub>	5

**Calorific Value.** The calorific value of a coal is a measure of its total heating power. It may be expressed as the number of British Thermal Units evolved when one pound of the coal is completely burned (or the number of calories per gram). The calorific value of fuels will be discussed in Chapter V.

**Ultimate Composition.** In addition to the determinations of moisture, ash, volatile matter and fixed carbon, which are grouped together under the term "proximate" analysis, the "ultimate" composition may be determined.

This involves determinations of the major elements present in coal. Carbon, hydrogen, nitrogen and sulphur are determined by analysis, oxygen being ascertained by difference. Phosphorus and chlorine may also be determined. These determinations, unlike those of the "proximate" analysis, call for considerable chemical skill.

## TYPES OF COALS

In considering the various types of coal, peat must be mentioned since it may bear a resemblance to the progenitor of coals.

**Peat.** Peat in the bog or swamp may contain more than 90 per cent. of water. At the top it is light in colour, but at lower depths the peat becomes darker and finally nearly black, by which time it is not so obviously of vegetable origin. It is frequently low in ash, but for use as a fuel the water content presents a serious problem, and much money has been spent endeavouring to dry peat on a commercial scale. To meet local fuel requirements, hand cutting and drying in stacks has been practised for many centuries. There are extensive deposits in England and Scotland besides the more famous ones in Eire.

*Lignites and Brown Coal.* Lignites are intermediate in character between peat and normal coals. They are of great importance in many parts of the world, but occur only in small pockets in Great Britain, of which the best known is at Bovey Tracey. A number of varieties are recognised, namely, (1) woody or fibrous brown coal, (2) earthy brown lignite, compact but friable, (3) brown coal showing a cleavage and a slight woody structure, and (4) black lignite resembling true coal in appearance. Although given in this order it is not implied necessarily that these substances represent a sequence in the transition of peat-like substances to coal. They are all characterised by high moisture contents (30–50 per cent.) and break down readily to slack on weathering.

It would appear that no real attempt has been made to distinguish between lignite and brown coal. The dictionary defines lignite as “mineral coal retaining the texture of the wood from which it was formed.” It would therefore be reasonable to define lignite as a solid fuel more mature than peat, which shows distinct woody structure when examined by the naked eye or under a pocket lens, and to consider brown coal as being a still more mature fuel, and devoid of obvious woody structure. The lignite would have a higher moisture content (50 per cent.) and lower carbon content than the brown coals (moisture 30 per cent.).

*Bituminous Coals.* The term “bituminous coal” has been applied to coals varying, on ash-free dry material, from about 75 per cent. carbon to about 91 per cent. carbon.

As the carbon increases, the oxygen of the coal decreases. The volatile matter varies on dry ash-free coal from about 45 per cent. for a coal containing 75 per cent. of carbon down to about 23 per cent. for a coal containing 91 per cent. of carbon. The calorific value of the dry ash-free material increases with the carbon content.

All bituminous coals show a banded structure due to bright and dull coal arranged parallel to the bedding planes. These bands have been classified by Stopes into four types—vitrain, clarain, durain and fusain. Without entering into further details here it may be said that vitrain and clarain together constitute the bright parts of the coal, and durain the dull and harder coal, known as Hards or Splints. Durain, as a rule, has a higher ash content than bright coal from the same seam, but the ash usually fuses at a higher temperature. Fusain is the dull and friable material usually found in thin layers along the bedding planes, although occasionally bands are found. It is the material known to miners as “Mother-of-Coal.” The division of a seam into bright and dull coal, when the bands are thick enough to separate, is often of great importance industrially, since the two constituents may have appreciably different properties. For example, the dull coal (Hards) of the East Midlands coal-field is much favoured for locomotive firing. This is not infrequently due to the fact that the ash has a high fusion temperature.

*Semi-Bituminous Coals.* Coals intermediate between bituminous coals and anthracite vary in carbon content from 91 to 93 per cent., and in volatile matter from about 23 per cent. to 10 per cent. on dry ash-free coal.

*Anthracite.* Coals of more than 93 per cent. carbon and 10 per cent. or below of volatile matter, are termed anthracites. They have a high lustre and are sometimes graphitic in appearance, but evidence of banded structure can still be found by careful examination. Most anthracites occur in the older geological formations, although not all the older coals are anthracitic.

*Other Types.* There are several other coals which are regarded as distinct types. One of the more important of these is cannel. It is hard and particularly rich in plant remains so that the volatile matter may be as high as 56 per cent.

# INDUSTRIAL FUELS : COAL

## GROUPING OF COALS

The above description of coal types constitutes a rough grouping. Many attempts have been made to devise comprehensive classifications of coals in order that the properties and uses of a coal may be predicted from the analysis of a sample. In this country the classification due to Seyler has probably had the biggest vogue, but with the development of fuel technology, all classifications have a decreasing value, since wider ranges of coal can now be utilised than would have been possible a few years ago. For example, the varieties of mechanical stoker now employed in boiler practice enable many coals to be used for steam raising which were not so suitable previously. A coal can be obtained to suit a particular form of stoker, or a stoker installed to burn a particular type of coal. Developments in the gas-making industry, such as the use of blending and the development of vertical retorts, have added to the range of coals suitable for this purpose. Blending is also improving the value of many coals for coke manufacture. There is, therefore, an increasing tendency to rely upon a simplified classification, especially for bituminous coals. A classification of this nature is given in Table 1.

TABLE 1. CLASSIFICATION OF BITUMINOUS COALS

Description				
	Group 1	Group 2	Group 3	Group 4
	Freeburning or non-caking	Slightly caking	Medium caking	Strongly caking
Analysis of Dry Ash-Free Coal:—				
Carbon, per cent. . . . .	78-81	81-82.5	82.5-84	84-89
Hydrogen, „ „ . . . . .	5.1-5.6	5.2-5.6	5.2-5.6	4.5-5.5
Volatiles, „ „ . . . . .	45-41	42-38	39-35	37-25
Calorific value				
B.Th.U./lb. . . . .	13860-14400	14400-14670	14670-15030	15030-15660
cal./gm. . . . .	7700-8000	8000-8150	8150-8350	8350-8700
Inherent moisture, per cent. . . . .	16-10	10-7	7-4	4-2
Type of coke, Gray-King Assay	A B	C D E	F to G <sub>5</sub>	G <sub>2</sub> to G <sub>9</sub>
W-D Swelling test . . . . .	1	1½-2½	3-4½	5-9

It may prove desirable to divide group 4 into two groups, as follows:—

	Group 4	Group 5
Carbon, per cent. . . . .	84-86.5	86.5-89
Volatiles, „ „ . . . . .	37-32	32-25
Calorific value :		
B.Th.U./lb. . . . .	15030-15550	15550-15660
cal./gm. . . . .	8350-8650	8650-8700

Group 5 would contain coals used for making coke with a high shatter test and generally non-reactive.

It will be noted that the calorific value and the percentage of carbon of the dry ash-free coal, as well as the caking and swelling properties, increase in passing from Group 1 to Group 4. On the other hand the percentage of volatile matter and of inherent moisture decrease in the same order.

The majority of industrial bituminous coals fall into one or other of the groups. There are exceptions, as, for example, "Hards" which, in general, do not cake; there are also marginal cases. The main purpose of the grouping is to indicate certain properties which have an important bearing on industrial uses.

The distribution of these coals in the various coal-fields of England and Scotland is indicated in the following table, which should be considered as a preliminary guide pending more complete examination.

STATISTICS	THE COALS.
Derbyshire ..	Pits in extreme south.
Nottinghamshire ..	Pits in southern part of eastern area.
Lancashire ..	Top seams in a very few pits.
Northumberland ..	Upper seams in extreme north.
Scotland ..	Many coals in Fifeshire and the Lothians.

GROUP 2—COALS are often described as free-burning, but they have slight caking properties and are slightly swelling.

Derby and Notts ..	..	A large number of coals.
Yorkshire ..	..	The Shafton seam, and the Barnsley and High Hazel seam in the Doncaster area.
North Staffordshire ..	..	The Great Row seam, Woodhead seam, and many coals on the east side.
Lancashire ..	..	The upper seams such as Florida and Crombouke.
Northumberland ..	..	Many coals north of Backworth, especially upper seams.
Scotland ..	..	Many in Fifeshire, the Lothians and Ayrshire—also in Stirling and Lanark.

GROUP 3—COALS are decidedly caking, and swell moderately, and they include a number of coals which are used for gas-making and some coals used in coke ovens. If used in coke ovens they make what is known as "reactive" coke.

Derbyshire ..	..	Especially north-west area.
Nottinghamshire ..	..	Very few.
Yorkshire ..	..	Most seams in South Yorkshire except the Sheffield-Barnsley and Doncaster areas. Coals in West Yorkshire.
North Staffordshire ..	..	Most of the coals in the west.
Lancashire ..	..	Trencherbone downwards.
Northumberland ..	..	South of Backworth—also lower seams toward north.
Scotland ..	..	Certain seams in central area.

GROUP 4—COALS are strongly caking and highly swelling and are chiefly valued for their use in coke ovens to produce a hard and generally unreactive metallurgical coke. The lower end of the group includes the Durham gas coals.

Yorkshire ..	..	In the Sheffield-Barnsley area—also some in West Yorkshire.
North Staffordshire ..	..	A few seams in the west.
Lancashire ..	..	Lower Mountain Mine, and bottom seams such as the Arley.
Durham ..	..	Practically all coals.
Northumberland ..	..	Extreme south, lower seams.
Scotland ..	..	North Central area—Dumbarton, Lanark, Stirling.

## SOUTH WALES COALS

The scheme of classification put forward for bituminous coals does not apply to the South Wales coals since their volatile matter content and coking properties are different from those of coals mined in other parts of the country. Coals generally are found to fall on a band, known as the "coal band" when their hydrogen content is plotted against their carbon content, both being taken on the dry ash-free (or dry mineral-matter-free) basis. When the South Wales coals are thus treated the result is as given in Fig. 4. It will be seen that the carbon content of the coals on the dry mineral-matter-free basis extends from 84 per cent. to 94.5 per cent., and the hydrogen content on the same basis is from 5.5 per cent. to 2.5 per cent.

It must always be remembered that a classification based on the inherent chemical properties gives a general picture of the types of coal, but it ignores such factors as size and ash content which, frequently, are very important in the selection of coals for specific purposes.

A general classification of South Wales coals similar to that of Table 1 is given in Table 2.

The volatile matter content, corrected for CO<sub>2</sub> in the dry ash-free coal, is a reliable first guide to the general properties of most of the commercial grades of South Wales coals. It can be made the basis of a broad classification if due

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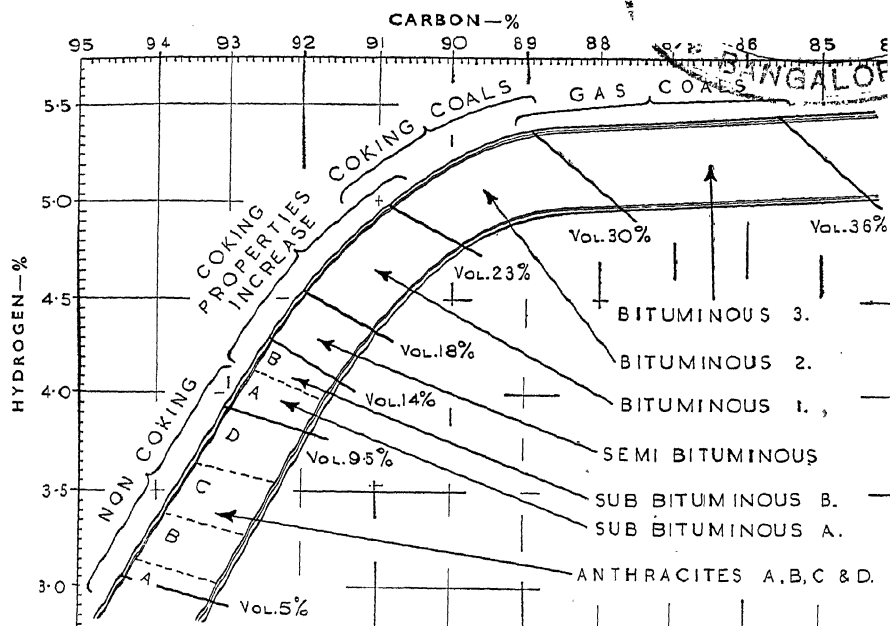


FIG. 4. The South Wales coal band. All quantities in this figure are on the dry mineral matter-free basis.

TABLE 2. CLASSIFICATION OF SOUTH WALES COALS

Description	Anthracite	Sub-bituminous	Semi-bituminous	Bituminous		
				1 Caking steam, coking, blending coals	2 Coking coals	3 Gas coals
	Non-caking	Non-caking to slightly caking	Slightly to medium caking	Caking coal	Strongly caking	Less strongly caking
Analysis of Dry Ash-Free Coal:—						
Carbon, per cent.	94.5-92	93-91	92.5-90.5	92-90	91-88	88-84
Hydrogen ..	2.8-3.9	3.9-4.25	4.0-4.5	4.3-5.0	4.8-5.3	5.0-5.5
Volatile matter	4.5-9.5	9-14	13-18	17-23	22-30	29-36
Calorific value						
B.Th.U./lb...	15250-15600	15600-15760	15800-15800	15650-15800	15800-15500	15500-15050
Cals./gm. ..	8472-8667	8667-8755	8667-8778	8694-8778	8778-8611	8611-8361
Inherent moisture, per cent.	1-3	0.6-1.0	0.6-1.0	0.6-1.0	1-2	1-3
Type of coke (Gray-King assay) ..	—	A B	G <sub>1</sub> , G <sub>2</sub> , G <sub>1</sub> , G	G <sub>9</sub> , G <sub>10</sub> , G	G <sub>10</sub>	G



regard is paid to other properties which are important for certain specific purposes. The volatile matter, however, can serve only as a pointer.

The proportion of volatile matter in South Wales coals ranges approximately between 5 and 36 per cent. The coals with the highest volatile content are found along the eastern, south-eastern and southern outcrop. The direction of maximum change in volatile matter is east to north-west, south-east to north-west, and south to north-west; the coals with the least volatile matter, the high grade anthracites, are found in the north-west corner of the coal-field, although small quantities are also mined in Pembrokeshire.

The coals of South Wales thus change profoundly in character and with considerable regularity as they are traced across the coal-field. This regional variation is always much greater than any change that occurs from seam to seam in the vertical succession at any one place. For this reason the place of origin of any particular coal is very significant for on it will depend the nature of the coal concerned. It thus arises that valleys, in which so much of the mining activity is concentrated, are frequently associated with certain commercial classes of coal.

The anthracites and coals up to 10 per cent. volatile matter (on the dry ash-free basis) are shown in Fig. 4 to be non-caking. The coals of 13.5 per cent. volatile matter have incipient caking properties, so that it can be taken that coals of from 10–13.5 per cent. volatile matter possess negligible caking properties.

Frequent reference is made to the dry steam coals of South Wales. There appears to be no generally accepted definition of these, but coals within the range 10–13.5 per cent. volatile matter are certainly of this class. Some might prefer to extend this range of volatile matter to 14 per cent., but at 14–14.5 per cent. the coals have measurable caking properties.

The following facts illustrate the rapid increase of caking properties as the volatile matter increases up to about 23 per cent. A coal of 16 per cent. volatile matter has well-defined caking properties. A coal of 18–19 per cent. volatile matter has marked caking properties. There is a further increase in caking properties between 19 and 23 per cent. volatile matter until a coal with 22 to 23 per cent. is used for making a first-class metallurgical coke.

Coals with more than 30 per cent. volatile matter show a slight decrease in caking power. There are no high-volatile, non-caking or feebly-caking coals in South Wales. The South Wales coals with more than 30 per cent. volatile matter are usually described as gas coals.

When comparing South Wales coals with coals from other coal-fields, certain factors should be noted. For instance, the free-burning coals of South Wales have, compared with the free-burning coals of elsewhere, a much lower percentage of volatile matter, a much lower percentage of inherent moisture, less hydrogen and a considerably greater calorific value on the dry ash-free basis. On the other hand, for combustion they require a greater draught.

With the exception of slight variations which only occur at the extreme ends of the range, all these coals have a high fairly constant calorific value and low inherent moisture.

Most of the South Wales coals have an ash fusion temperature of over 1,350° C. in an oxidising atmosphere and above 1,300° C. in a reducing atmosphere. Many have a much higher ash fusion temperature, but coals with less than 10 per cent. and greater than 30 per cent. volatile matter tend to have a lower ash fusion temperature than the others. It is, however, not uncommon for some of these to be quite high.

## THE COAL-FIELDS OF GREAT BRITAIN

Geological evidence suggests that the original vegetable debris was probably laid down in four large tracts (excluding the Kent field), but these have since

## INDUSTRIAL FUELS : COAL

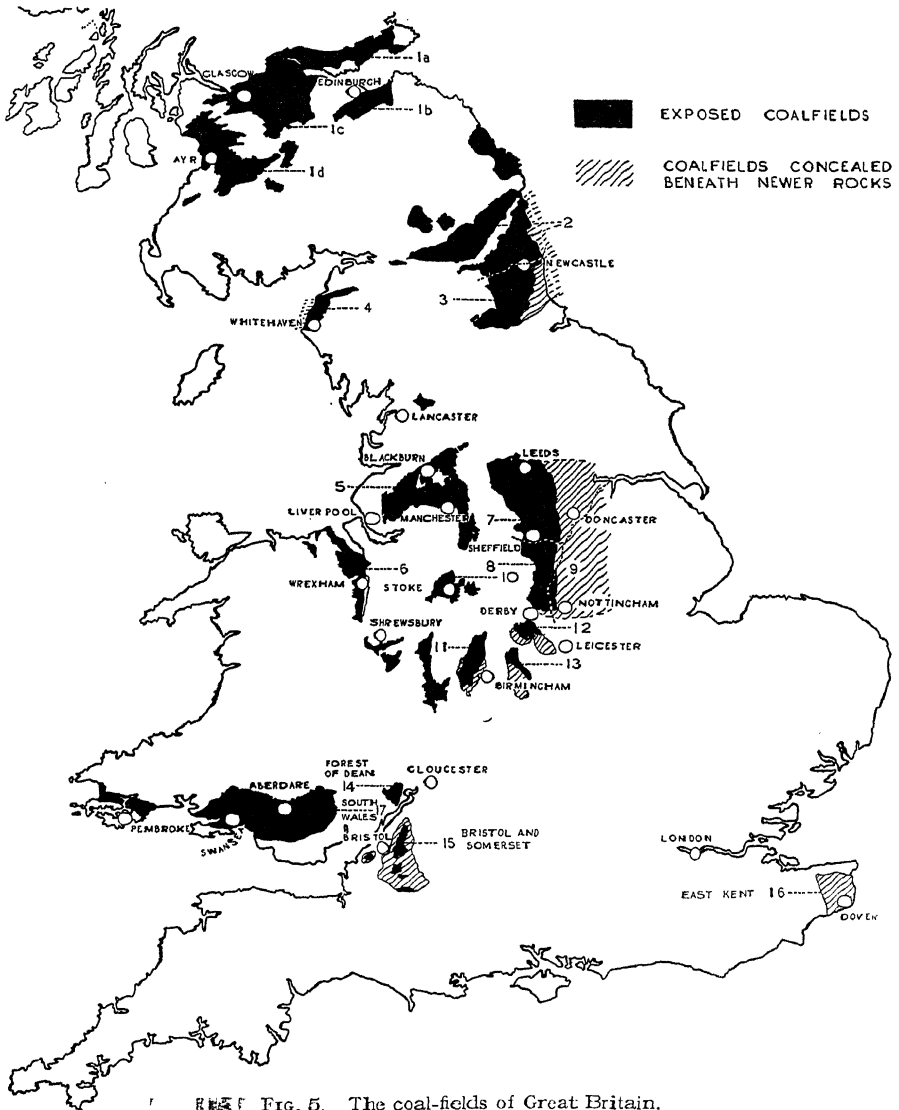


FIG. 5. The coal-fields of Great Britain.

1. Scotland.
2. Northumberland.
3. Durham.
4. Cumberland.
5. Lancashire and Cheshire.
6. North Wales.
7. Yorkshire.
8. Derbyshire.
9. Nottinghamshire.
10. North Staffordshire.
11. South Staffordshire (and Cannock Chase).
12. Leicestershire and South Derbyshire.
13. Warwickshire.
14. Forest of Dean.
15. Bristol and Somerset.
16. East Kent.
17. South Wales.

become separated into the forty detached areas now known, of which only twenty have any importance. These four areas were :

- (1) Scotland.
- (2) Northumberland, Durham and Cumberland.

(3) North Wales, Lancashire, Yorkshire and the Midlands.

(4) South Wales, Forest of Dean and the Bristol and Somerset coal-fields.

The distribution of the British coal-fields is shown in Fig. 5. The exposed coal-fields, where coal measures are actually at the surface, are shown in black. The concealed coal-fields, that is, those coal measures which are covered by later rocks but where the coals are within workable distance of the surface, are shaded.

A survey of the coal-fields of the country undertaken by the Fuel Research Board has been in progress for many years. Details of this survey are given in Appendix 1.

Table 3 gives the size and outputs of the coal-fields and indicates in some measure the types of coal produced in each field. This information is to be regarded as a preliminary guide pending the preparation of more detailed data. The groups mentioned refer to the classification in Table 1, except for South Wales.

TABLE 3. THE COAL-FIELDS OF GREAT BRITAIN

District	Approx. area. sq. miles	Output in 1938. Million tons	Types of coal produced
Scotland .. ..	1,700	30.3	Freeburning; Group 1: in Fifeshire and the Lothians. Slightly caking; Group 2: in Fifeshire, the Lothians and Ayrshire and also in Stirling and Lanark. Medium caking; Group 3: Certain seams in the central area (Lanark). Strongly caking; Group 4: North Central area—Dumbarton, Stirling and Lanark. Pseudo-anthracite: Lanark, Fifeshire, Stirling.
Northumberland ..	800 with Durham	13.3	Group 1: A few in extreme north of area—upper seams. Group 2: Most of Northumberland and north of Backworth, especially upper seams. Group 3: South of Backworth; also lower seams toward north. Group 4: Extreme south, lower seams.
Durham .. ..	See above	31.4	Group 4: Practically all Durham coals belong to Group 4.
Cumberland ..	150	1.6	Mostly Groups 3 and 4.
Lancashire and Cheshire	500	14.3	Group 1: Top seams in a few pits—very occasionally. Group 2: Upper seams such as Florida and Crombouke. Group 3: Lower seams such as Trencherbone. Group 4: Lower Mountain Mine seams—bottom seams, as Arley in many places.
North Wales ..	150	3.7	Mostly Group 3. Some border on Group 4.
Yorkshire .. ..	2,136 with Derby and Notts	42.4	Group 2: Shafton seam and Barnsley High Hazel seams in Doncaster area. Group 3: Coals in West Yorkshire district, most seams in South Yorkshire except in the Sheffield-Barnsley and Doncaster areas. Group 4: Coals in the Sheffield-Barnsley area; also some coals in West Yorkshire.

District	Approx. area sq. miles	Output in 1938 million tons	Types of coal produced
Derbyshire ..	See above	13.4	Group 1 : In extreme south. Group 2 : A large number of coals would probably be of this group. Group 3 : North-west area.
Nottinghamshire	See above	15.4	Group 1 : Southern part of eastern area. Group 2 : Most Notts. coals would probably be in this group. Group 3 : Very few.
North Staffordshire	100	6.9	Group 2 : Great Row and Woodhead seams; many coals on eastern side. Group 3 : Most of the coals on the west. Some might be classed as Group 4.
South Staffs. and Cannock Chase	150	6.6	Group 1 : All coals although one or two border on Group 2.
Leicester and South Derbyshire	84	3.6	Group 1 : All coals.
Warwickshire ..	60	5.6	Group 1 : All coals.
Forest of Dean ..	35	1.3	Mainly Group 3.
Bristol and Somerset	240	0.8	Mainly Group 4. Some border on Group 3.
East Kent ..	206	1.8	Group 4 : Low-volatile steam coal.
South Wales ..	1,000	35.3	See special table of South Wales coals (Table 2) (5.5 million tons of anthracite).

## THE MAIN USES OF COAL

At this point it is appropriate to consider the principal characteristics, apart from size, required in coals for their main uses.

*Coals for Steam Raising.* Formerly it was considered that the type of coal required for steam raising needed to be free-burning, and the term "steam coal" was in general use. Since the development of forced draught, mechanical stokers and modern boiler plant, the range of coal for steam raising has been greatly widened, and every type of coal, from free-burning to strongly caking, is being used. With further developments it will, in future, be possible to burn all types of coal satisfactorily for steam raising, but in some existing plants, particularly where hand firing is practised and draught is limited, coals of Groups 1 and 2 will give the best results.

Small, hand fired boilers are inclined to smoke badly when fed with high-volatile coals, and for such boilers, low-volatile coals are more suitable if adequate draught is available.

The evolution of volatile matter depends upon the nature of the coal and affects its behaviour on the grate. High-volatile coals of Group 1 release their volatile matter rapidly, giving smoky flames and low firebed temperatures. Group 3 coals do not release their volatiles so rapidly, and the gases evolved are richer in hydrocarbons. The less rapid release of volatiles permits them to be burned with a better-regulated supply of air and thus to give hotter flames and a high firebed temperature. The high firebed temperature may be due to the less reactive type of coke produced from good coking coals, which thus burn to  $\text{CO}_2$  and form little CO in the fuel bed. These hot coals, if used with a

low ash content, may damage the links or firebars. The character of the fixed carbon, which varies considerably between the groups, has much to do with the temperature generated in the firebed with a given air supply.

*Coking Coals.* The essential property of a coking coal is that it will produce a strong hard coke suitable for metallurgical purposes; this is best obtained from coals which combine a high caking power with a relatively low percentage of volatile matter. The coals which stand out pre-eminently are the coking coals of Durham and South Wales (carbon about 88 per cent. and volatile about 23-30 per cent. on the dry ash-free coal). Reserves of these coals, however, are not large, but it has been proved possible to obtain satisfactory cokes by using blends of coals which hitherto had been considered quite unsuited for coke production. The use of narrower coke ovens has also assisted in widening the choice of coal. Thus coals used to produce coke now include a number of coals formerly used only for gas manufacture or for household purposes. The term "coking coal" is therefore becoming more catholic in its application.

For metallurgical purposes the coke must be low in sulphur and phosphorus, and this restriction removes a number of otherwise suitable coals from consideration.

*Gas-Making Coals.* For gas making in horizontal retorts it is desirable that the coal used shall form a good coke and at the same time have a high volatile content (Groups 3 and 4) to ensure a high yield of gas. With the advent of continuous vertical retorts and the steaming of the charge, however, the range of coals which can be successfully carbonised has widened in the direction of lower caking power (Group 2), particularly when sized coals can be used. The term "gas coal" has not, therefore, the same significance as formerly.

*Manufacturing Coals for other than Steam Raising Purposes.* There are many types of furnace and kiln used in industry, and the design of the installation usually decides which type of coal is the most suitable. Generally a coal of moderately high volatile matter (Groups 1 and 2—Table 1) is required for furnaces of a reverberatory type, but where a high local temperature is required, lower-volatile coals such as Group 3 are more suited, but these need more draught. For producer gas manufacture the best fuels are the non-caking or weakly-caking coals of Groups 1 and 2. The fusion temperature of the ash should be reasonably high unless the producer is of the slagging type.

Where the flue gases from burning coal may come into contact with food-stuffs, attention must be paid to the amount of certain elements, such as arsenic, which may, more rarely, be present in the coal in quantities high enough to be technically significant.

## THE CHARACTERISTICS OF COAL OF INDUSTRIAL IMPORTANCE

There has been given above a brief description of the properties of the various types of coal available. It is now necessary to enter into more detail of the evaluation of a coal for industrial purposes. For most practical purposes it is sufficient to have a knowledge of four characteristics:—

- (1) The quantity of inherent moisture.
- (2) The quantity of volatile matter.
- (3) The caking properties of the coal.
- (4) The quantity of ash, and its fusion temperature.

*The Inherent Moisture.* The quantity of inherent moisture present in a coal affords considerable information upon the general character of a coal as will be seen from Tables 1 and 2. Coals of high rank, i.e. of high carbon content and low volatile matter, have little inherent moisture (0.6 to 2 per cent.), and, roughly speaking, as the volatile matter increases, so does the inherent moisture. Good caking coals may have inherent moisture contents of from 1 to 4 per cent.

At about 5 to 7 per cent. of inherent moisture the coals are feebly caking, and the non-caking bituminous coals contain some 10-16 per cent. of inherent moisture.

*The Volatile Matter.* The volatile matter based on dry ash-free coal affords confirmatory evidence of the type of coal, particularly in the low-volatile range below 20 per cent. Between 20 and 30 per cent. volatile matter, good coking coals are obtained. Above 30 per cent. the value of the volatile matter is less precise in fixing the caking properties of the coal.

Roughly speaking the length of the flame obtained during combustion varies as the volatile matter. Coals with high percentages of volatile matter (Groups 1 and 2) burn with long flames (long-flaming coals), and low-volatile coals with short flames (short-flaming coals)—Groups 3 and 4.

A reverberatory type of furnace or some types of kiln require a long-flaming coal since it is desirable to obtain heat away from the grate. For intense local heating, a short-flaming coal must be employed. In boiler practice coals high in volatile matter are useful where sudden increases in load are anticipated.

Closely connected with volatile matter is the rate of combustion of a coal. Low-volatile coals are slow burning, and high-volatile coals are fast burning. The high-volatile coal evolves a considerable proportion of its weight as gas, tar and water, and the resulting gas and tar can be burned above the grate more rapidly than a corresponding weight of solid fuel on the grate, although special attention must be paid to the combustion of the gas and tar in order to avoid smoke and to obtain complete combustion.

The residue left on heating a non-caking coal is usually more reactive than the coke yielded by a caking coal. That is to say, it will combine more readily with oxygen to give carbon dioxide, or react with carbon dioxide to give carbon monoxide. A caking coal and a non-caking coal show, therefore, very different behaviour on a grate. The coke from the caking coal burns more slowly than the residue from non-caking coal and gives a higher fuel bed temperature. Moreover, a caking slack will form larger pieces of coke having a much smaller external surface area per unit volume, and thus burning more slowly.

The volatile matter has also an important effect on ignitability since it is the volatile matter which ignites first. Distillation of volatile matter precedes combustion of the fuel. Hence low-volatile coals, for example anthracites, are difficult to ignite whereas high-volatile coals ignite readily.

*The Caking of Coal.* The caking properties and swelling power constitute probably the most important characteristic of coals.

The reasons leading to the variations in the caking properties of coals are not fully understood. Theories have been advanced to correlate the caking of coals with the amount of extract which the coals yield when treated with solvents such as pyridine or with benzene under pressure. Although it is true that strongly-caking coals yield a larger proportion of certain of these extracts than non-caking coals, it has become clear that the non-extractable portion of the coal also plays an important part in the caking process. The theories connecting amount of extract with caking power do not completely explain the marked improvement in the coke arising from carrying out the coking under increased pressure.

The mechanism of coke formation depends on the facts that (1) caking coals soften to greater or less extent and become plastic during a certain temperature range, (2) fluid material is extruded on to the surface of the particles and grains of coal and (3) a pressure is set up in the coking mass due to gas which forces the fluid materials to cover the surfaces of the pieces of coal where they act as a cement.

It is found that when a caking coal, crushed into a mass of smaller pieces (e.g. slack) is so heated that it is raised in temperature at a constant rate

of some  $1^{\circ}\text{C.}$  to  $3^{\circ}\text{C.}$  per minute, the sequence of events is as follows (see Fig. 6) :—

No visible change occurs up to about  $300^{\circ}\text{C.}$  ( $512^{\circ}\text{F.}$ ), but at about this temperature the evolution of considerable amounts of oxides of carbon and water vapour shows that the coal is beginning to be decomposed. If a stream of inert gas be passed through the heated coal it is found that at about  $370^{\circ}\text{C.}$  ( $698^{\circ}\text{F.}$ ) the coal begins to offer a decided resistance to the passage of the gas. This temperature depends on the rank of the coal and is higher as the volatile content of the coal decreases; it may be over  $400^{\circ}\text{C.}$  with a low-volatile bituminous coking coal. Individual particles of coal begin to swell and careful examination reveals that certain constituents of the coal have begun to fuse. Over this temperature range, in addition, small quantities of oils are evolved from the coal.

From about  $370^{\circ}\text{C.}$  ( $698^{\circ}\text{F.}$ ) upwards, the resistance of the coal to the

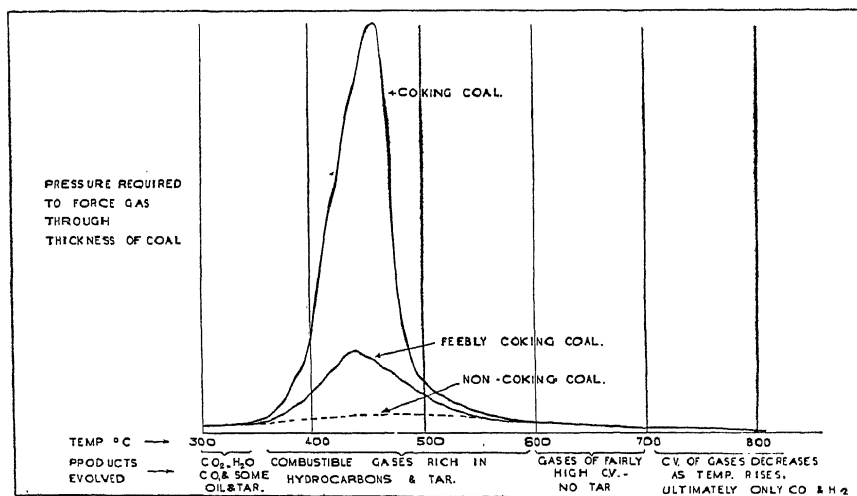


FIG. 6. The behaviour of coal on heating.

passage of gas increases very rapidly, and in a good caking coal, if the coal was originally in the form of slack, it is found that at about  $430^{\circ}\text{C.}$  ( $806^{\circ}\text{F.}$ ) a very considerable pressure is needed to force gas through it. During this period a certain amount of gas and tar continue to be evolved. The coking mass is in the pasty condition which is known as the "plastic state." At a somewhat higher temperature—frequently between  $440^{\circ}\text{C.}$  and  $450^{\circ}\text{C.}$  ( $824^{\circ}\text{F.}$ – $842^{\circ}\text{F.}$ )—the resistance of the coal to the passage of gas rapidly decreases to a very low value. At this stage the plastic mass has hardened into semi-coke. Semi-coke is not plastic and there is now little further resistance. Quantities of gas and tar which have been imprisoned in the plastic mass are now released. The presence of this gas in the plastic coal has previously caused a high internal pressure, and the quality of the coke produced from the coal is dependent to a considerable degree upon this pressure. As before, the temperatures mentioned depend on the volatile content of the coal, being with a low-volatile coal higher than those here mentioned.

Tar continues to be evolved up to a temperature of about  $600^{\circ}\text{C.}$  ( $1,112^{\circ}\text{F.}$ ) accompanied by considerable quantities of gas rich in hydrocarbons and of a

high calorific value. It is during this stage that most of the benzole-forming constituents are evolved from the coal; the benzole is not formed as such, but is the product of the secondary decomposition, at a higher temperature, of the primary bodies formed in carbonisation.

As the temperature increases to 700°–900° C. (1,292°–1,652° F.) the quantity of gas produced decreases, and this gas is of lower calorific value. The gas now consists mainly of carbon monoxide and hydrogen.

For coals with a lower percentage of volatile matter, the temperature range of plasticity increases. Thus, for a coal with 30–35 per cent. volatile matter, it may be 370°–450° C. (698°–842° F.), whereas for a coal with 22 per cent. volatile matter, the plastic range may be 420°–530° C. (788°–986° F.). As the coals used become less strongly caking the resistance to the passage of gas during this temperature range becomes less until, with a non-caking coal, there is, of course, no increased resistance since these coals do not pass through a plastic phase.

In practice, the caking and swelling properties of coal may be greatly dependent on the size of the coal, on its percentage of ash, on the rate of burning or gasification, on the method of firing and on the fuel bed thickness. These properties may also be affected by weathering or by heating in store. These will be briefly discussed.

(a) *The Size of Coal.* The phenomena of caking and swelling are greatly modified by the size of the individual pieces of the coal. Whereas a small piece of caking coal of the order of  $\frac{1}{8}$  inch diameter will tend to swell and become plastic as a whole upon heating through the plastic range (370°–450° C.) (698°–842° F.), and will, therefore, easily cohere to the neighbouring particles, a larger piece of coal, say of the order of  $\frac{1}{2}$  inch and over, will not swell in this way, but fluid matter will be forced out through the sides along the bedding planes to form blisters of fused material on the surface. With this behaviour there is less opportunity for the individual pieces of coal to become fused together in such a way as to form massive lumps of coke such as are formed when slack is used. These lumps present such a small relative surface that it is difficult to get effective contact between air and coal. It is thus often found that whereas a coal, by reason of its caking properties, will give considerable difficulty in a furnace when charged in the form of slack, it can readily be used when in the form of nuts or doubles. It will also be found that whereas a slack of medium caking properties (Group 3) may be too caking for the purpose for which it is used, a coal in the form of singles or nuts from the more strongly caking Group 4 may give satisfaction on account of its size, and in spite of its stronger inherent caking properties. Generally the slack from a coal is more strongly caking than the larger material, even when the larger material has been crushed to the same size as the slack.

(b) *The Amount of Ash.* The amount of ash has a material influence on caking properties. Under certain conditions a medium or fairly strongly-caking slack, containing 5 per cent. of ash, may be more difficult to burn on account of strong coke formation than the slack from the same coal containing 15 per cent. of ash. This reduction of caking power by non-caking matter has suggested the blending of coke breeze and anthracite duff with caking coals to modify their burning characteristics. At least 10 per cent. anthracite duff can be used in this way when the caking index of the basic coal (its "agglutinating value") is not lower than 15 when determined by the British Standard method, corresponding generally to group 4 coals (Table 1).

(c) *The Rate of Burning or Gasification.* Experience with chain-grate stokers has shown that a high rate of burning per square foot of grate area reduces the caking properties of coals, and the formation of large pieces of coke, and leads to better combustion conditions.



(d) *The Method of Firing, and the Fuel Bed Thickness.* A free-burning or medium-caking slack, burned in a thin bed, may give unsatisfactory results through an excessive amount of fines falling through the grate, and through excessive grit emission. The same coal burned on a stoker of the retort or coking type using a thick fuel bed will give more satisfactory results in this respect, because the type of heating tends to increase the caking properties and the fines are thus trapped in the fuel bed.

On the other hand a strongly-caking slack will form massive lumps of coke on the coking type of stoker and be difficult to burn, whereas, spread out in a thin layer, the fluid plastic material will be burned away before masses of coke can be formed.

### SULPHUR IN COAL

Sulphur is present in coal in general to the extent of 0.5 to 2.5 per cent., although occasionally coals are met with containing much larger amounts. It is present in the original coal in three forms, namely as pyrites ( $\text{FeS}_2$ ), as organic sulphur compounds and as sulphates.

Under the action of heat the pyritic sulphur is partly evolved in the form of free sulphur according to the reaction  $\text{FeS}_2 = \text{FeS} + \text{S}$ ; this reaction occurs at about  $500^\circ \text{C}$ . ( $932^\circ \text{F}$ ). The remaining sulphur left in the form of  $\text{FeS}$  may behave in any or all of three ways according to the conditions. In the presence of ample supplies of oxygen it may be oxidised according to the reaction  $4\text{FeS} + 7\text{O}_2 = 2\text{Fe}_2\text{O}_3 + 4\text{SO}_2$ . The ferrous sulphide may in itself combine with some of the other constituents of the ash forming ferrous silicate with the liberation of sulphur; this reaction assists the formation of clinker, as ferrous silicates have a low melting point. Finally, a proportion of the sulphur first liberated may combine with the carbon, forming organic sulphur compounds.

The organic sulphur compounds are burned to  $\text{SO}_2$  in the course of combustion.

At high temperatures sulphur present as sulphate is partly removed from the coals in the form of sulphur trioxide. Sulphur is thus objectionable for the following reasons:—

- (1) Oxides of sulphur contaminate the atmosphere, forming sulphurous and sulphuric acids which cause serious damage to property.
- (2) Part of the sulphur may be absorbed by inter-action with the grate bars, and since sulphide of iron is comparatively fusible, it may give rise to serious trouble.
- (3) When the furnace is used for metallurgical purposes the sulphur may pass into the metal under treatment.
- (4) When a fuel containing hydrogen is burned, it forms water vapour. If the temperature of the gases falls too low, for example on the surface of the economiser, some of the water is condensed, and the oxides of sulphur are absorbed by this water, forming a solution of sulphurous and sulphuric acids. This may be the cause of severe external corrosion of the metal. The presence of  $\text{SO}_3$  in flue gases has been shown to raise the dew-point of the gases materially and thus to increase the danger of deposition of acids on the metal.

Iron pyrites is a contributory cause of clinker formation, but can be removed to a great extent by appropriate cleaning methods.

### COAL PREPARATION AND CLEANING

The coal as mined, termed "run-of-mine," comprises a mixture of sizes, from the finest particles of slack less than  $\frac{1}{32}$  inch diameter up to very large lumps of 2 or 3 feet. It also includes dirt unavoidably collected from the roof

or floor in the process of mining, or occurring as thin dirt bands in the coal itself. The shale may vary from almost pure stone to a carbonaceous material containing 60 per cent. or more of combustible matter.

The material is first screened in order to separate the large coal. The large material may be over 3 inches or 4 inches in size according to local practice. The large coal is passed over picking belts where the dirt is removed by hand. The purity of this coal, which is used principally for domestic purposes and locomotives, depends upon the efficiency of the hand picking.

The small coal is generally washed and sold for industrial purposes. In 1927, 20 per cent. of the coal was cleaned in 500 washeries and 27 dry cleaning plants. In 1934, 40 per cent. of the coal was cleaned in 611 washeries and 150 dry cleaning plants. In 1938, the proportion of coal cleaned had risen to 45 per cent. of the total output. Since much of the large coal is hand-picked, the proportion of the coal that is cleaned is a good deal higher than these figures indicate.

The operation of coal cleaning processes depends on the relationship between

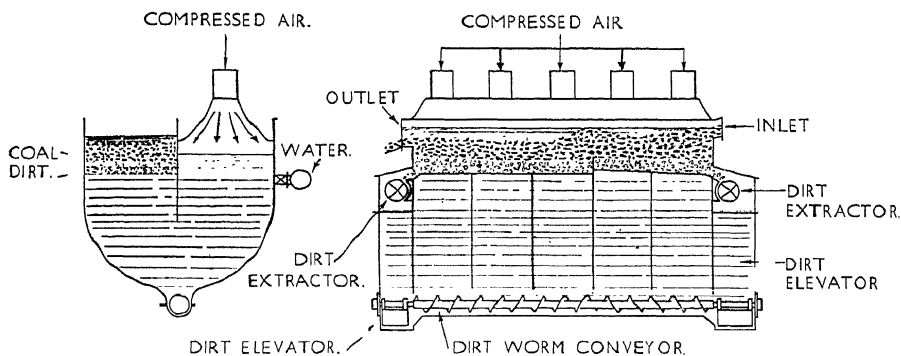


FIG. 7. The Baum coal washer.

the velocity of fall of the particle in water and its density. Expressed mathematically :—

$$V = C\sqrt{d(s-w)}$$

$V$  = velocity of fall in water, ft./sec.

$d$  = diameter of particle (inches).

$(s-w)$  = diff. in sp. gr. of material and washing medium (water = 1).

$C$  = constant, about 1.8 for nodular pieces.

The specific gravity,  $s$ , of the pure coal is between 1.25 and 1.4, of the shale 2.5, and of pyrites, 5. Thus, for a constant diameter of particle, i.e. a constant value of  $d$ , pyrites and shale will fall to the bottom of a tank of water more rapidly than pure coal. If the water be given an up-and-down motion, the upward velocity of the water can be made sufficiently high to cause the coal then to travel upward while the shale and pyrites continually travel downward. Fig. 7, showing a Baum washer, gives an illustration of a practical form of washing plant. A pulsating motion set up by compressed air causes the water to move up and down in the coal bed with the necessary velocity, and thus keeps the dirt at the bottom of the bed and the coal at the top. A fast-flowing stream of water carries the raw material into the wash-box, and the clean coal is carried out of the box, over a suitable gate to the drainage screens. A gate at the outlet end of the wash-box is lifted periodically by the washery attendant to remove the dirt, or can be operated automatically. The pulsating

motion was formerly applied by a plunger acting in the water in the plunger box. This arrangement is still found on older plants.

The velocity of fall, however, also depends upon the size of the particle,  $d$ . In the form of purely relative figures it can be calculated that:—

$$\begin{array}{ccccccc} \text{Where } d = & \frac{1}{16}'' & \frac{1}{8}'' & \frac{1}{4}'' & \frac{1}{2}'' & & \\ V = & 0.09 & 0.12 & 0.18 & 0.35 & 0.5 & 0.71 \end{array}$$

The shape of the particle has also an important bearing on the rate of fall, a flat piece of shale will fall with different motion and generally with a faster velocity than a rounded piece of the same gravity.

In some types of washing plant it is necessary to screen the coal into a number of sizes before washing. In others, however, in order to reduce the amount of plant necessary, to diminish the labour costs and to simplify the procedure as much as possible, normal practice is to treat the unscreened coal in one operation and, after washing, to screen out the slack below about  $\frac{1}{8}$  inch. This fine material may be re-washed.

Under these conditions it is impossible to avoid a small proportion of dirt being mixed with the coal, but, in general, a good washer should not leave more than 2 per cent. of removable dirt in the coal.

If the value of  $w$ , the specific gravity of the liquid in which the coal is placed, exceeds the specific gravity of the pure coal, the coal will float. Ingenious practical methods have been devised for cleaning coal in this way. Recently, practical processes have been evolved using suspensions of sand, clay (loess), barytes or other finely divided solids in water as a washing medium of the correct specific gravity. Also a solution of calcium chloride, with a specific gravity of 1.4 has been employed, and very clean coal has been obtained.

The increasing development of processes employing such suspensions appears likely to lead to the production of considerable quantities of low-ash coal in future years.

Other types of coal washing plant are operated with an upward current of water, such that the coal is floated out of the main stream and recovered, while the impurities sink and are rejected.

Another highly important type of plant is that of dry cleaning. In this the material does not come into contact with water so that the ash removed is not, even in part, replaced by water mechanically held. Dry cleaning plants utilise an upward current of air introduced beneath a pulsating table. Raised parallel riffles cause the clean coal, the high-ash coal, and the dirt each to flow to different parts of the table edge where they are recovered separately.

In coal washing the fines present a problem of no little difficulty. In dry cleaning methods they are given off from the coal as a cloud of dust in the air current from which they have to be separated by mechanical devices. In wet washing they form a slurry with the water that is very difficult to settle. Methods have now, however, been devised whereby more rapid settlement can be effected. The settled slurry is high in ash. The formation of slurry can be prevented to a considerable extent by dedusting the coal before wet cleaning.

The distribution of dirt in the raw coal coming from the pits may be very variable according to the friability of the shale. Where the pieces of dirt are separate from the coal there is no particular difficulty in separation, but it often happens that where there are dirt partings in the seam, small pieces adhere to the coal, and the combined specific gravity is sufficiently low for the particle to be classified as coal. A certain amount of ash inevitably arises from this cause, and this forms in the coal washing process an intermediate product known as "middlings." The intermediate product may also be due to a close inter-mixture of the dirt with the coal so that separation of dirt and coal cannot be effected unless the particles are crushed to a fine powder. Finally,

## INDUSTRIAL FUELS : COAL

in some instances, the coal and dirt are so intimately associated that separation of the two is not possible by any known means.

It will be recognised, therefore, that the operation of coal washing must be carried out in a skilled manner. Excess ash in coal may be due to practical difficulties of operation at the washery or to the output of the colliery under war conditions being so great that the existing washery is over-loaded. Some collieries are not equipped with coal cleaning plant.

### THE SIZE OF COALS

The smaller coals below 3-4 inches, separated from the first screening, are generally (although not invariably) washed. They may be sold without further treatment, but usually are screened into sizes for industrial uses. They may be divided into two broad classes :—

- (1) Sized or graded coal.
- (2) Slacks.

*Sized Coals.* Sized coals are those which have been screened between two sizes and from which the greater part of the fines has been removed. They are known by such names as pearls, peas, beans, singles, nuts, etc., but these names are used unsystematically, the beans of one colliery often being equivalent in size to the singles or nuts of another colliery. The real criterion of size should be a size analysis test where a sample is taken and passed through B.S.I. standard square mesh sieves. Such tests show the results given in Table 4 for sized industrial coals :—

TABLE 4. GRADING OF COMMERCIAL SIZES OF COALS

	$\frac{1}{2}$ –0 inch per cent.	$\frac{1}{2}$ – $\frac{1}{4}$ inch per cent.	1– $\frac{1}{2}$ inch per cent.	1 $\frac{1}{2}$ –1 inch per cent.
Pearls .. ..	40 30	55 64	5 6	— —
Peas .. ..	8 10	78 65	14 25	— —
Beans .. ..	8 6	53 41	39 53	— —
Singles .. ..	6 4	30 18	60 72	4 6

There is, however, no sharp line of demarcation between the groups. In view of the importance now attached to the influence of size on combustion and gasification rates and efficiency, it is desirable that, in future, precise information of the above nature should be used rather than that reliance should be placed on trade names which may be misleading.

*Slacks.* In general, slack may be defined as that material which has been screened through one upper screen and contains, with certain exceptions to be described later, all the fine coal.

The upper size may be as big as 2 inches ; the slack may then be almost as useful as a graded fuel ; or the upper size may be as low as  $\frac{3}{8}$  inch when the slack is often difficult to handle if wet.

Slacks are known in various districts by different names, such as dross, smalls, duff, gum, etc. These names do not guarantee any specific size, and, as stated above a size analysis should be made. According to the size of the

aperture in the upper screen, the size grading of slacks should be approximately as in Table 5.

TABLE 5. SIZE GRADING OF SLACKS

	$\frac{1}{2}$ -0 inch per cent.	$\frac{1}{2}$ - $\frac{1}{4}$ inch per cent.	$\frac{1}{4}$ - $\frac{1}{8}$ inch per cent.	$1$ - $\frac{1}{2}$ inch per cent.	Over 1 inch per cent.
2 inch slacks ..	20	14	20	24	22
$1\frac{1}{2}$ inch .. ..	22	16	23	27	12
$1\frac{1}{4}$ inch .. ..	26	20	26	25	3
1 inch .. ..	30	21	30	19	—
$\frac{3}{4}$ inch .. ..	41	29	27	3	—
$\frac{1}{2}$ inch .. ..	45	39	16	—	—
$\frac{3}{8}$ inch .. ..	52	40	8	—	—
South Wales ..	52	15	16	12	5
do. ..	35	14	15	18	13

The size distribution of South Wales slacks may differ from that of slacks in the rest of Great Britain because, whereas it is customary in the rest of Great Britain to size coals through screens perforated by round, square or other shaped holes, in South Wales it is often graded over parallel bars set  $1\frac{1}{4}$  inches apart, which pass all the fines and also much larger pieces. This gives an entirely different size distribution.

*The Size of Coal: General.* Partly owing to the failure to use size analysis there is a lack of reliable information about the relationship between size and combustion and gasification efficiencies, and draught requirements.

The size of coal should be such that the lumps present enough surface for good combustion but are not so small as to prevent easy penetration of air. Uniform grading (or sizing) is a valuable asset in that it produces a fuel bed which offers even resistance to the passage of air through it, and therefore will result in uniform combustion or gasification.

It must not be forgotten, however, that the size of particle of the fuel when it is carbonised or partly carbonised may be altered by its caking properties, causing particles to cohere.

Sized coal avoids segregation. In a slack which consists of relatively large particles together with fine dust, the larger coal frequently becomes concentrated in one zone of the hopper or heap, with the result that the mechanical stoker may at one time be fired with nutty material and at another time with little more than dust, or, as frequently occurs on chain-grate stokers, the sides of the stoker may receive all the nutty material, and the centre the dust. The result is an uneven fuel bed resistance and indifferent combustion conditions. Care should be taken to mix the coal thoroughly when waggons of dry slack are unloaded since it is usual for the centre of the waggons to contain a greater proportion of fines than the sides or ends. If the centre of the waggon is discharged first, then for several hours the combustion appliance will be using fuel with a high percentage of fines. Later on it will receive much coarser material drawn from the ends of the waggon.

Many of these, and other, difficulties would be avoided if all the fines were removed from coal and used in powdered fuel installations, and only "sized" coal were marketed for general industrial purposes.

## CHAPTER II

### INDUSTRIAL FUELS : FUELS DERIVED FROM COAL

Carbonisation—Coke—Coal gas (Town gas)—Producer gas—Blast furnace gas—Water gas—Coal tar fuel oils—Electricity.

**F**UELS derived from coal comprise coke, coal gas, producer gas, water gas, blast furnace gas, and oil fuels derived from the distillation of the tar produced in coal carbonisation. Electricity may also be regarded as falling in this category. Many of these fuels are obtained by the carbonisation of coal. Large quantities of mixed gas (coke oven gas and blast furnace gas) are used in the heavy industries, where both of these gases are by-products.

All coals do not when carbonised produce coke as a commercial product, a fact which has been made clear in the preceding chapter. For industrial carbonisation therefore, it is at present necessary to confine ourselves to certain classes of coal. Some coals are highly coking and the slack from these is carbonised in coke ovens for the production of hard metallurgical coke. These coals generally yield from 35 per cent. to 22 per cent. of volatile matter as carbonised in practice, determined on dry coal containing about 5 per cent. of ash. Coals towards the higher-volatile end of this group and caking coals of still higher-volatile content generally produce a high gas yield and a slightly weaker coke and are used in the gas industry.

Coking coals are limited in quantity and within a comparatively short time this country will face a real shortage of first quality coking coals. It is of the utmost importance, therefore, that the best use should be made of the coal that we possess, and that wherever possible the best coking coals should be reserved for metallurgical purposes. Research is in progress which it is hoped may lead to a commercial method of manufacturing town gas of the present quality from any coal irrespective of its coking qualities, but this process even if fully successful is not likely to become commercially available for some years.

In coal carbonisation the coal is heated in a closed chamber from which air is excluded, the heat being transmitted to the coal through walls constructed of fire brick or silica brick. A portion of the gas or coke obtained in carbonisation is used to produce the necessary heat.

The gases are led from the carbonising chamber through mains and are first cooled in order to remove as much as possible of the tar and water vapour evolved from the coal; the remaining tar is removed from the gases by a mechanical process, the gases being then washed by water or sulphuric acid to remove ammonia from which sulphate of ammonia is manufactured. Benzole is extracted from the gases by washing with oil or by other methods, the oil being subsequently distilled to recover the benzole which is then purified for use. The residual gases comprise coal gas which according to its origin and use is variously styled coke oven gas or town gas. There is no essential difference between these gases. Where the gas is to be used for town purposes and also for many industrial purposes it is purified from hydrogen sulphide by being passed over iron oxide.

The coke which remains in the retort at a temperature of perhaps 1,000° C. is discharged, cooled, e.g. by being sprayed with water, by heat interchange with the incoming secondary air, or by means of cold flue gas free from oxygen (dry-cooling), and is then sent to the coke preparation plant where it is graded into sizes. A proportion of it is generally broken and re-sized for special purposes.

## COKE

The yield of coke from coal in practice varies from 60 per cent. to 80 per cent. of the weight of the coal charged into the retort. It thus represents both by weight and thermal value the principal product of coal carbonisation.

Coke consists essentially of carbon and mineral matter accompanied by some 1 to 2 per cent. of sulphur and small quantities of hydrogen, nitrogen and phosphorus. When marketed, gas coke should contain less than 6 per cent. of water and metallurgical coke less than half that quantity. It is possible to calculate the calorific value of coke within sufficiently close limits for most industrial purposes from a knowledge of the ash and moisture contents and on the assumption that the remainder of the coke consists of carbon mostly in a graphitic form. Anticipating the information to be given in Chapter V the calorific value of coke in B.Th.U per lb. can be calculated from the expression:—

$$(\text{carbon per cent.} - \text{ash per cent.} - \text{water per cent.}) \times 14,200 \div 100 \\ \text{for oven coke}$$

$$\text{it.} - \text{ash per cent.} - \text{water per cent.}) \times 14,400 \div 100 \\ \text{for gas coke.}$$

The composition of a typical dry ash-free coke is:—

Carbon .. .. .	95 per cent.
Hydrogen .. .. .	1 „
Nitrogen and sulphur .. .. .	2 „
Oxygen (and errors) .. .. .	2 „

Coke possesses very little volatile matter, this being rarely over 4 per cent. and generally from 2 to 3 per cent. as derived from gas works. Many gas cokes and most metallurgical cokes contain less than 2 per cent. of volatile matter. This volatile matter is evolved almost wholly as carbon monoxide and hydrogen.

An important property of the coke is its resistance to handling, a property often termed its "hardness." By the nature of its method of manufacture and from its subsequent quenching with water, coke is strained mechanically and may contain a number of small cracks. Upon handling or while in use these cracks may cause the coke to break and to form smalls. The diminution in the size of coke is deleterious for many purposes though not for others. The test most commonly used for measuring the strength of coke is the shatter test. In this test 50 lb. of lump coke over 2 inches in size is dropped four times from a height of 6 feet on to a cast iron or steel plate and the amount of breakage is measured by screening the coke. Table 6 gives typical figures:—

TABLE 6

	Gas retort coke	Metallurgical coke			Low temperature coke
Shatter test:		South Wales	Yorkshire	Derby	
Over 2 inch, per cent.	76	87.5	78	57	73
„ 1½ inch „	88	95	88	72	76
Through ¾ in. „	2	1	2	3	14

A good metallurgical coke should show 75 per cent. over 2 inches, 85 per cent. over 1½ inches, and less than 3 per cent. below ¾ inch size. This test has been found by experience to be of great value in selecting cokes for specific purposes. A hard coke, for example, is a good hot melting coke.

A metallurgical coke is dense whereas most gas cokes are more porous. A

high porosity decreases the crushing strength of the coke and increases the rate of reaction with oxygen to some extent. Where a very high temperature is desired a dense hard coke burned with mechanical draught will give the best results. Where a less high temperature is needed an open, porous and softer coke will prove quite satisfactory and can be burned with less draught.

The size of cokes has an important bearing on their combustion properties. On small appliances particularly, a reduction in size may compensate for low combustibility. The size of coke should be selected to suit the appliance in

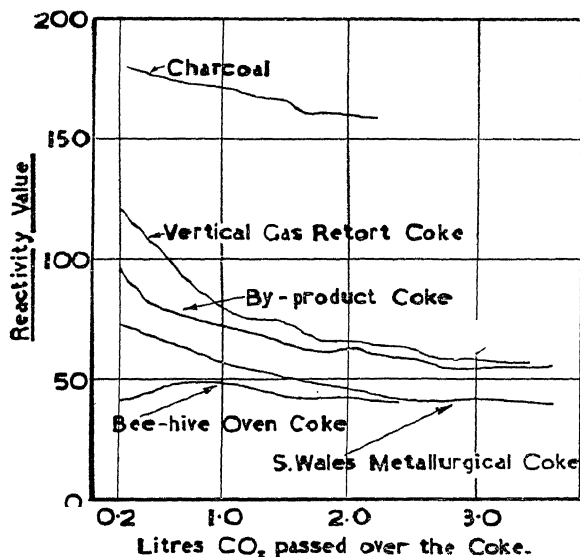


FIG. 7A. Curves showing the reactivity of several cokes and charcoal, determined by passing carbon dioxide over the material heated to 950° C. (Fuel Research Station). The reactivity as measured by the conversion of the carbon dioxide to carbon monoxide will be seen to decrease as the material is heated for a longer time and more carbon dioxide is passed over it, until it reaches a virtually constant value.

which it is used. The standard sizes (square mesh screens) for gas coke are as follows :—

- No. 1. No upper limit over  $1\frac{3}{4}$  inches.
- 1a. Within the limits  $3\frac{1}{2}$  inches to  $1\frac{3}{4}$  inches.
2. Within the limits 2 inches to 1 inch.
3. Within the limits  $1\frac{1}{4}$  inches to  $\frac{1}{2}$  inch.
4. Within the limits  $\frac{3}{4}$  inch to  $\frac{3}{8}$  inch.
5. Unwashed fuel approximately  $\frac{3}{4}$  inch to  $\frac{1}{4}$  inch from which the fines up to  $\frac{1}{8}$  inch have been removed.
- Breeze. Generally  $\frac{1}{2}$  inch to 0.

The local coke associations (or the gas companies) will advise on which size of coke should be selected for a particular purpose.

The temperature at which coke ignites depends on the temperature to which it has been subjected during the process of manufacture, and upon the conditions in the retort. Some cokes will ignite at temperatures as low as 450° C. and according to the conditions under which they have been made the temperature will increase until a hard coke made by the old beehive process that is used for foundry work may not ignite until well over 600° C. These tem-



peratures are somewhat arbitrary, however, since they depend to some extent upon the experimental conditions under which the determination was made.

Reactivity or combustibility is another important property of coke. The stronger the coking power of the coal from which the coke is produced and the higher the temperature at which it has been carbonised the less easily will the coke react with oxygen or carbon dioxide. Reactivity is generally measured by the rate at which a coke will convert carbon dioxide into carbon monoxide at a given temperature and under standard conditions. Differences in this respect between different fuels are shown in Fig. 7A.

The reactivity of coke is not only a function of its physical properties, but is also increased by the presence of reducible iron compounds, of sodium carbonate, of lime, and of certain other inorganic substances. A highly reactive coke will give a lower fuel bed temperature than a coke of low reactivity.

The ash content of coke is entirely governed by the mineral matter content of the coal from which it is made. The carbonisation industry endeavours to produce coke having as little ash as possible, but the economics of coal cleaning place a limit to the degree to which it is at present considered desirable to clean coals. Most cokes have between 5 and 12 per cent. of ash, but figures outside these limits both in the lower and higher directions are sometimes found.

The water content of coke depends upon the manner in which it is cooled. Water is sprayed on the hot coke, mechanically or by hand, in quantity sufficient to cool it so that it will not ignite when standing in air or when loaded into waggons. If the right amount of water is used, the residual heat in the interior of the lumps will evaporate the surface moisture and the coke will be virtually dry. With too much water, cooling the coke too far, the water content of the finished coke may be excessive. Mechanical quenching, being under better control, generally results in lower average moisture content over a period than hand quenching. Where water is not used, as in "dry quenching," the coke is dry. A little moisture in coke is not disadvantageous.

Sulphur is a deleterious element in coke for certain purposes, particularly for metallurgical uses. For ordinary combustion purposes sulphur has little influence unless the gases derived from the combustion of coke or from the producer gas made from coke are to come in contact with goods in a furnace which would be detrimentally affected by sulphur dioxide. Any sulphur present in the coal as iron pyrites ( $\text{FeS}_2$ ) is first converted into iron sulphide ( $\text{FeS}$ ) at a temperature of some  $500^\circ\text{C}$ ., and at higher temperatures part of the ferrous sulphide will react with hydrogen forming some metallic iron and liberating hydrogen sulphide. Most of the sulphur present in the form of  $\text{FeS}$ , however, remains in that form and thus appears in the coke. Sulphur present as sulphates may be distilled from the coal, though generally the amount of sulphates in a coking coal is very small indeed. Sulphur present in the coal as organic sulphur is partly retained in the coke and is partly evolved in the form of carbon disulphide and other organic sulphur compounds which are to be found in coal gas. Similarly a part at least of the sulphur distilled from pyrites reacts with the coke to form carbon disulphide ( $\text{CS}_2$ ), while part is now known to interact with the coal to form organic sulphur that is retained in the coke. Generally the sulphur content of cokes ranges from 0.75 to 1.5 per cent.

The quantity of phosphorus present in cokes varies between 0.002 per cent. and 0.025 per cent. in this country, but some may contain as much as 0.3 to 0.5 per cent. Phosphorus is only important when the coke is required for metallurgical uses.

Coke has a true specific gravity of 1.75 to 2, this being the specific gravity of the material without pores. The "apparent" specific gravity, i.e. the specific gravity of the lump of coke with the pores varies from 0.75 to 1. The porosity is generally between 40 and 60 per cent.

Table 7 gives some general information upon the bulk density of coal and coke broken for use and having a normal content of moisture and ash.

TABLE 7

Fuel				Bulk density Pounds per cubic foot	
Anthracite	..	..	..	55-60	
Bituminous coal	..	..	..	50-55	
Steam coal	..	..	..	50-55	
				$2-1\frac{1}{2}$ inch	Rough
Coke oven coke	..	..	..	26-30	29-32
Horizontal retort coke	..	..	..	23-26	25-28
Vertical retort coke	..	..	..	21-22	23

*Evaporative Value of Coke.* Tests which have been carried out to determine the net evaporative value of coke prove that 1 lb. of coke of average good quality has in a boiler a net evaporative value of 9 to 10 lb. of water per lb. of coke, from and at 212° F.

*Combustion Rate.* In ordinary circumstances combustion rates of 12 lb. to 16 lb. per square foot of grate per hour can be maintained with natural draught. With a suitable forced draught furnace a good average rate of combustion is 25 lb. per square foot of grate per hour, although it is sometimes possible to increase this to as high as 35 lb. to 40 lb. per square foot of grate per hour. The best results are usually obtained under conditions of balanced draught.

## COAL GAS

The composition and calorific value of coal gas depends upon the method by which it is made. In coke ovens and horizontal gas retorts the gas produced is of high calorific value and consists of the coal gas without any admixture. Removal of benzole and purification may bring the gas down to the calorific value at which it is supplied to the public without further treatment. Some times, however, producer gas or water gas is added to it in order to reduce the calorific value.

The calorific value of town gas is controlled by Act of Parliament. Through the Ministry of Fuel and Power, gas companies make a statutory declaration of the calorific value in B.Th.U. per cubic foot gross at which they will supply gas. Tests are made independently of the gas companies to ascertain that at no time does the calorific value fall below the declared standard. Coke oven gas is supplied to domestic and industrial purchasers through the gas undertakings according to the powers given to them by Act of Parliament.

In continuous vertical retorts a certain amount of steam is blown in from the bottom of the retort achieving the double objective of partly quenching the coke and of being itself converted into water gas through its passage up the retort by the reaction:  $C + H_2O = CO + H_2$ . Since water gas consists essentially of carbon monoxide and hydrogen, the different methods of carbonisation influence the percentage of these two gases in the ultimate gaseous mixture. The typical composition of a number of forms of coal gas is as in Table 8.

Town gas, whether derived from coke ovens or made from coal at a gas works, has the same general composition and characteristics. The gas companies supply gas of a declared calorific value which is generally between 40

TABLE 8. COMPOSITION OF VARIOUS TYPES OF COAL GAS

	Horizontal retorts	Continuous vertical retorts without steaming	Continuous vertical retorts steamed	Debenzolised coke oven gas
Oxygen, $O_2$ .. ..	0.4	0.4	0.4	0.4
Carbon dioxide, $CO_2$ .. ..	2	3	4	2
Hydrocarbons, " $C_mH_n$ " .. ..	3.6	3	2	2.6
Carbon monoxide, CO .. ..	8	9	18	7.4
Hydrogen, $H_2$ .. ..	52	53.6	49.4	54.0
Methane, $CH_4$ .. ..	30	25	20	28
Nitrogen, $N_2$ .. ..	4	6	6.2	5.6
Calorific value— B.Th.U./cu. ft.				
Gross .. ..	560	540	475	525
Net .. ..	500	482	426	467
Sp. gr. (air = 1) .. ..	0.42	0.43	0.48	0.38

and 500 B.Th.U. per cubic foot. In 1940, of 654 gas undertakings in the country that had a declared calorific value, 219, including some of the biggest companies in the country, operated at 500 B.Th.U. per cubic foot; 119 between 475 and 495; and 223 between 450 and 470 B.Th.U. The consumer is charged by the therm, a therm being 100,000 B.Th.U. The number of cubic feet of gas comprising a therm is given by the fraction

$$\frac{100,000}{\text{declared C.V.}}$$

The consumer is thus charged on the therm basis for the actual gross amount of heat with which he is supplied. The declared value of a gas is always based on the gross C.V. of the gas (see Chapter V).

The calorific value of any mixture of gases is the sum of the calorific values of the constituents. The calorific value of town gas can thus be calculated by adding the calorific value of its content of  $H_2$ ,  $CH_4$ , CO,  $C_mH_n$ , the oxygen,  $CO_2$  and  $N_2$  having no calorific value. The uncertainty that arises from making this calculation in town gas is that the methane generally contains a small proportion of ethane which has a higher C.V. than methane, and more particularly that the composition of the hydrocarbons designed  $C_mH_n$  depends on the method of manufacture. According to data contained in British Standard publications the several gross calorific values are as follows at 60° F. and 30 inch bar. saturated with water vapour:—

Carbon monoxide, CO .. ..	318 B.Th.U. per cubic foot
Hydrogen, $H_2$ .. ..	320
Methane, $CH_4$ .. ..	
Ethane, $C_2H_6$ .. ..	
Ethylene, $C_2H_4$ .. ..	
$C_mH_n$ (unstripped horizontal retort and coke oven gas) .. ..	
(unstripped vertical retort gas)	3,040
(for benzole-stripped gas) .. ..	1,560

The cost of gas per gross therm as received is considerably greater than the cost for solid fuels on the same basis. There are, however, reasons why town gas is proving of great value in industry and why its use in industry is increasing (see Chapter XXXIV).

The comparative freedom from sulphur of town gas is a matter of con-

## INDUSTRIAL FUELS : FUELS DERIVED FROM COAL

siderable importance in many heating operations matter will be further discussed in Chapter XVIII. in the retorts and after washing for ammonia and ben. some 500-700 grains of sulphur per 100 cubic feet, or even  $\frac{1}{2}$  than this, the sulphur being principally present in the form of hydrogen sulphide. By extraction of hydrogen sulphide by means of iron oxide, the total sulphur content of the gas is reduced to some 20-30 grains per 100 cubic feet. Even this small quantity, however, is sometimes deleterious and the present trend of practice in many large gas companies is to reduce the sulphur content of a gas to less than 10 grains per 100 cubic feet. To effect this further reduction special plant is necessary, this being either a modification of the benzole extraction process or involving a catalytic process.

Coal gas cannot be highly preheated on account of its hydrocarbon content. The higher hydrocarbons,  $C_mH_n$ , and to a lesser extent methane, decompose into carbon and hydrogen at a red heat. The air for combustion can, however, be preheated. A coal gas generally requires some  $4\frac{1}{2}$  to 5 volumes of air per volume of gas so that if the air for combustion is preheated the fact that the gas is not preheated has comparatively little influence on the final temperature reached.

### PRODUCER GAS

The method of manufacturing producer gas is discussed in a later chapter in this book. Here it is sufficient to state that it is manufactured from coal or coke by blowing a mixture of steam and air through a bed of incandescent fuel, generally not less than 3 feet in depth. The gases leaving the fuel bed consist of a mixture of carbon monoxide and hydrogen accompanied by all the nitrogen in the incoming air. When coal is used as the fuel, distillation of volatile matter in the upper portions of the fuel bed causes the producer gas to be accompanied by the tar and gas liberated from the coal. As the temperature of the gases at this point may be comparatively high a large part of the tar is retained in the vapour state, whilst the remainder is carried mechanically in the form of minute drops of "tar fog."

The characteristics of producer gas will therefore depend upon whether the gases are made from coke or from coal, and upon whether it is used in the hot crude state or whether it is cooled and cleaned. Typical compositions of cooled, clean gas are as in Table 9.

TABLE 9. COMPOSITION OF COOL, TAR-FREE PRODUCER GAS

	Bituminous coal		Coke	Anthracite
	Static producers	Mechanical producers		
Composition, percent. by volume				
Carbon dioxide, CO <sub>2</sub> .. ..	4-7	4-7	4-6	4-6
Carbon monoxide, CO .. ..	20-28	22-29	24-28	25-27
Hydrogen, H <sub>2</sub> .. ..	10-13	10-16	10-12	14-18
Hydrocarbons, C <sub>m</sub> H <sub>n</sub> .. ..	0.3-0.7	0.3-0.7	Nil	About 0.4
Methane, CH <sub>4</sub> .. ..	2.8-3.5	2.8-3.3	0.4-0.6	1.2-1.3
Nitrogen, N <sub>2</sub> .. ..	52-57	52-57	52-55	50-53
C.V.—B.Th.U/cu. ft. at 60°F. and 30 inch bar.				
Gross .. ..	140-160	140-160	130-140	132-140
Net .. ..	134-152	134-152	125-134	
Sp. gr., air = 1; (approx.) ..	0.9	0.9	0.9	

If the gases are derived from bituminous coal and are not cooled or cleaned the additional quantities of tar contained in them may increase the calorific value by some 15 B.Th.U. per cubic foot gross. Gases derived from coke and anthracite contain no appreciable quantities of tar.

If the gases are not cooled they contain in addition to their heat of combustion sensible heat which adds to the heat developed in the furnace on combustion. The sensible heat of the gases between 60° F. and 1,470° F. (15°–800° C.) is some 17 B.Th.U. per cubic foot.

Producer gas is characterised by a large quantity of inerts ( $\text{CO}_2 + \text{N}_2$ ) and consequently by a low calorific value. It requires, however, considerably less air for its combustion than town gas, approximately equal volumes of producer gas and air being required. Thus a theoretical mixture of gas and air required for combustion, containing no excess air, has for coal gas a gross calorific value of 94–95 B.Th.U. per cubic foot, whilst for producer gas the calorific value is between 65 and 72 B.Th.U. per cubic foot. By using the producer gas without allowing it to cool a good deal of this difference can be made up and of course the overall efficiency of gas production can be increased. If the air is also preheated producer gas can give substantially the same furnace temperature as coal gas.

Producer gas is generally not purified from sulphur for industrial heating purposes because of the cost of purifying so large a volume of gas. Most of the sulphur contained in the coal will be found in producer gas generally in the form of hydrogen sulphide, but accompanied by small quantities of sulphur dioxide.

For furnace heating by coals containing appreciable quantities of volatile matter, by oil and by coal gas the composition of the air-gas mixture fed to the furnace and the setting of the primary and secondary air inlets is exceedingly important. Much of the efficiency of utilisation depends on the correct setting of the primary air controls and the secondary air dampers. When using producer gas at temperatures up to about 1,000° C. it is generally advisable to close the air control at the injector and thus to use no primary air.

The uses of producer gas are limited by:—

- (a) Its comparatively low flame temperature (see Chapter V).
- (b) The presence of dust.
- (c) The presence of sulphur compounds.

TABLE 9A. PRINCIPAL DIFFERENCES BETWEEN PRODUCER GAS AND TOWN GAS

Property	Producer gas	Town gas
(1) Calorific value B.Th.U./cu. ft. (gross), 60° F., 30 inches mercury, saturated	130–160	450–500
(2) Flame temperature, deg. C. (approx.)	1,600	2,200
(3) Dust, oz./1,000 cu. ft.	0.02–0.03	Nil
(4) Sulphur grains/therm with $\text{H}_2\text{S}$ removed	10–15	2–6
Sulphur grains/therm no $\text{H}_2\text{S}$ removed	40–55	(All $\text{H}_2\text{S}$ must be removed as a Statutory requirement)
(5) Specific gravity (air = 1)	0.9	0.42–0.48
(6) Air required for complete combustion cu. ft./cu. ft.	0.9–1.2	4.25–4.5
(7) Hydrogen content, per cent.	14–18	45–55

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### GAS

gas is rarely used outside the iron and steel industry and com-  
will therefore be included here upon this subject. When iron

is smelted with coke in the blast furnace a blast of air is introduced through the tuyeres at the bottom of the furnace which burns the coke to carbon monoxide. Part of the carbon monoxide reacts with iron ore in the furnace forming  $\text{CO}_2$ . The gases issuing from the top of a blast furnace are therefore similar to producer gas made without the addition of steam, but higher in  $\text{CO}_2$  content. The calorific value of blast furnace gas is from 90–105 B.Th.U. per cubic foot, and the gas customarily contains about 11 per cent. of  $\text{CO}_2$ , 27 per cent. of  $\text{CO}$ , 2 per cent. of  $\text{H}_2$  and 60 per cent. of  $\text{N}_2$ .

The calorific value of the theoretically completely combustible air-gas mixture is only 54 B.Th.U. per cubic foot, and for the production of a high temperature it is thus necessary to preheat both gas and air.

Blast furnace gas can be used under proper conditions quite successfully for industrial heating and for gas engines. It contains a great deal of dust from which it can be purified by appropriate methods. For gas engine work and for heating coke ovens the dust must be removed from the gas almost completely. As an illustration of the potentialities it may be mentioned that like producer gas it is used successfully for heating coke ovens operating with a flue temperature of 1,800°–1,400° C., both gas and air being preheated to 1,000° C.

## WATER GAS

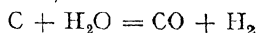
Water gas is principally used by gas undertakings as a method of making town gas, complementary to carbonisation. When water gas is distributed by a gas undertaking, it is always mixed with coal gas. There are comparatively few works outside the gas industry that manufacture water gas; a small number of engineering works use it for such processes as water gas welding, it has been used for heating metallurgical furnaces, and in the chemical industry it finds certain limited applications principally as a source of hydrogen. It seems probable that it will find more applications in the chemical industry in the future.

What is known as “blue” water gas, because it burns with a blue flame, is produced by the action of steam on carbon at a high temperature, the gases produced being carbon monoxide and hydrogen, accompanied by small quantities of carbon dioxide and nitrogen. The range of composition is as follows:—

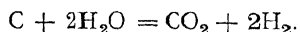
	Per cent.					
Hydrogen .. .. .	..	..	..	..	..	45–51
Carbon monoxide .. .. .	..	..	..	..	..	40–45
Carbon dioxide .. .. .	..	..	..	..	..	3–6
Nitrogen .. .. .	..	..	..	..	..	3–7
Methane .. .. .	..	..	..	..	..	0.1–0.5

The calorific value is about 290–300 B.Th.U. per cubic foot gross, and the net value some 20 B.Th.U. lower.

The solid fuel, coke or anthracite, is raised to incandescence by blowing the ignited fuel with air, the blow gases being allowed to escape to atmosphere, after giving up their heat to waste heat boilers. When the fuel bed is sufficiently hot, the air blast is discontinued, and steam is introduced in its stead. The steam is decomposed in contact with the carbon according to the chemical equation (Chapter IV):—



at somewhat lower temperatures a reaction sets in which forms carbon dioxide,



Both these reactions absorb heat and when the amount of  $\text{CO}_2$  in the gases becomes too high because the fuel bed has been cooled, the steam is discontinued and the air blast is restarted. A typical cycle is as follows:—

- (1) Blow with air for one minute, gases rejected to atmosphere.
- (2) Discontinue air blast, open steam valve, and after allowing a few seconds to remove the blow gas from the generator, open gas valve to scrubber, the gas being now made for use, for four minutes. Sometimes steam is blown up the fuel bed for one minute, down for two minutes, and up for one minute.
- (3) Steam shut off, stack valve and air blast opened, and the sequence repeated.

For town gas purposes, a second chamber is added filled with chequer firebrick-work. This chamber is maintained at a high temperature and into it, during part of the time when steam is being blown into the fuel bed, gas oil is injected. The gas oil is decomposed and vaporised yielding a gas rich in hydrocarbons. Some of these hydrocarbons condense forming water gas tar, but those which are permanently vapourised into the gas increase its calorific value considerably. When the blue gas has been carburetted, for example by the addition of some 7-8 per cent. of hydrocarbons, its C.V. may be raised to about 500 B.Th.U. per cubic foot. This gas is known as carburetted water gas.

### COAL TAR FUELS

The subject of the production, characteristics and use of coal tar fuels is discussed more exhaustively in this chapter than are other fuels. References to other fuels will be found scattered through this book, often in special chapters devoted to their utilisation, but apart from a little general information in Chapter XXVIII no specific reference is again made to coal tar fuels.

(i) *Sources of Coal Tar.* In Great Britain, crude coal tar is produced mainly as a by-product of the carbonisation of coal at high temperatures. A relatively small amount is produced by the various processes of low temperature carbonisation at present being operated.

The yield of crude tar varies with the nature of the coal and with the type of carbonising plant and the operating conditions.

By high temperature carbonisation, gas works vertical retorts produce 12-14 gallons of tar per ton of coal, gas works horizontal retorts 9-11 gallons and coke ovens about 8.5 gallons. The average amount is about 10 gallons, which is equivalent to a little more than 1 cwt., or approximately 5 per cent. by weight of the coal carbonised.

Low temperature processes yield 15-22 gallons of tar per ton of coal carbonised.

The total annual production of crude coal tar is of the order of 2,000,000 tons, of which approximately 60 per cent. is derived from gas works and 40 per cent. from coke ovens. Practically the whole make is distilled for the production of refined products. These include a range of materials which may be used as liquid fuels and a product suitable for use as a pulverised fuel.

(ii) *Nomenclature of Coal Tar Fuels.* It is possible to manufacture coal tar liquid fuels with kinematic viscosities (cf. Chapter IX), at ordinary temperature (60° F.), within the range 0.1 to 10<sup>1.2</sup> stokes. In practice, the differences in viscosity will mean that the various fuels must be heated to different temperatures for their viscosities to fall to the value required for efficient atomisation (0.25 stokes). There is thus established an approximate scale of equi-viscous temperatures which can be introduced into a scheme of nomenclature for the fuels. According to the handbook on the subject of coal tar fuels prepared by the Association of Tar Distillers, there is available a range of coal tar liquid fuels known respectively as Coal Tar Fuel 50, Coal Tar Fuel 100, Coal Tar Fuel 200, Coal Tar Fuel 250, Coal Tar Fuel 300 and Coal Tar Fuel 400. The figures represent the approximate temperature in degrees Fahrenheit at which the particular fuel is in a condition suitable for atomisation. It is the intention,

in this section, to give an outline of the derivation and general characteristics of these fuels. For further information relating to their properties and the technique of their application, the handbook mentioned should be consulted. General information on the use of oil fuel will be found in Chapter XXVIII.

(iii) *Derivation and Characteristics of Coal Tar Fuels.* The extent to which crude tar is distilled is governed by varying market requirements, particularly for the heavier products such as the coal tar fuels. Fig. 8 gives, in outline, the general scheme of tar distillation, showing the composition of the fuels. Special arrangements are generally made for distilling off the aqueous ammoniacal liquor and part of the light oil, after which the water-free tar is put through the main distillation plant.

A further quantity of light oil is then obtained, followed, in order of increasing boiling point, by the fractions known as middle oil, creosote oil and anthracene

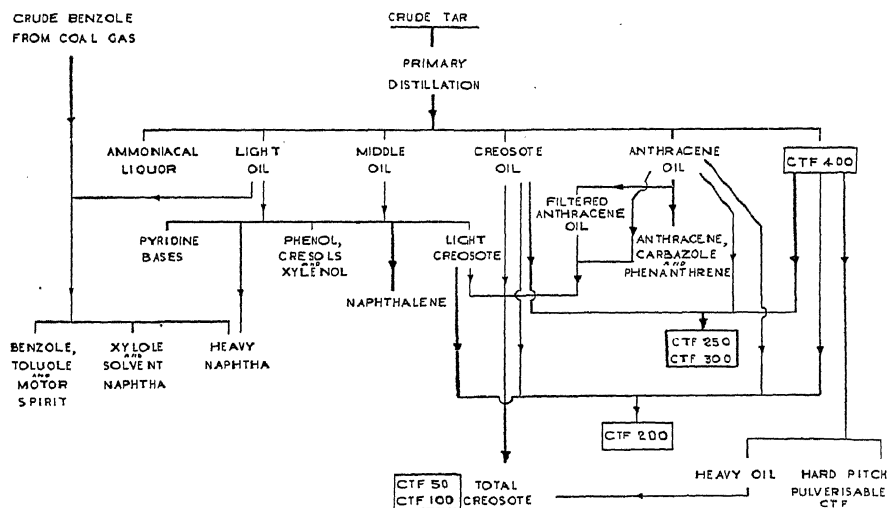


FIG. 8. Tar distillation.

oil respectively. The product remaining in the still is medium soft pitch or Coal Tar Fuel 400. It is generally run off into beds or bays where it is allowed to cool and solidify. It can then be excavated for delivery to consumers in lump form. In practice, it is generally used as a liquid fuel which means that if the lump fuel is received melting plant will be required. The process of melting and handling this fuel is comparatively simple. In certain circumstances this material is supplied in liquid form at a temperature of 300° F. and over, in loads up to about 12 tons. For this purpose, heat-insulated road tanks are employed and deliveries are usually restricted to a radius of about fifty miles from the tar works.

The distillation of tar normally finishes with the production of medium soft pitch. If a *Pulverisable Coal Tar Fuel* or hard pitch is required it is necessary to distil a further quantity of heavy oil from the pitch until a hard and brittle product is obtained which will pulverise readily at ordinary temperature. This part of the distillation process requires experience and special care.

The creosote of commerce is derived from several fractions. Creosote oil and anthracene oil are the main constituents, coupled with the light creosote remaining after the removal of phenols, naphthalene, etc., from the light and



middle oils. The heavy oil obtained during the manufacture of the hard pitch can also be included.

*Coal Tar Fuel 50* and *Coal Tar Fuel 100* are among the several grades of creosote produced by the careful blending of those oils. They distil above about 206° C. (392° F.) and contain at least 99.5 per cent. of material soluble in toluole. C.T.F. 50 is fluid down to 32° F., but is only available for special purposes in certain districts. At ordinary temperatures C.T.F. 100 generally contains a quantity of separated solids consisting mainly of naphthalene, but at temperatures above 90° F. it is completely fluid. It is available throughout the whole country.

Creosote—Grade A, and Creosote—Grade B, as marketed during war-time by the Petroleum Board, are identical with C.T.F. 50 and C.T.F. 100 respectively.

C.T.F. 250 and C.T.F. 300 may be regarded as a combination of medium soft pitch and anthracene oil, with varying proportions of the creosote oil fraction. This is generally achieved by stopping the distillation of the tar at a point during the collection of the creosote oil fraction when the product in the still conforms to the specification for *Coal Tar Fuel 300*. For this reason it is often referred to as a straight-run product. *Coal Tar Fuel 250* is prepared by the addition of creosote oil to C.T.F. 300 until the correct viscosity is attained. This is known as the "oiling-back" process.

If, in the oiling-back process, a suitably increased proportion of creosote is used, *Coal Tar Fuel 200* will be formed. In the diagram this fuel is shown as a combination of medium soft pitch with all the available oil fractions, but in point of fact it is necessary to distil an additional quantity of tar to obtain the amount of oil required. This is a disadvantage to the distiller and may, in ordinary times, affect the availability of this fuel.

C.T.F. 200 is similar to the Creosote/Pitch Mixture now marketed by the Petroleum Board.

The delivery of C.T.F. 400 has already been discussed. C.T.F. 300 and C.T.F. 250 are best transported hot in heat-insulated road tanks which have a delivery radius of about fifty miles. They are both viscous tarry products, and if packed in smaller, unheated or uninsulated containers, will be difficult to handle when cold. C.T.F. 200 is a much thinner material. Although thicker than an ordinary paint it is much thinner, for instance, than Golden Syrup. It can be transported by loading warm into road or rail tanks or by barge. C.T.F. 100 can be delivered warm in road tanks, but if rail transport is used, the tanks must be fitted with coils so that the fuel can be warmed by steam on arrival and any separated solids (naphthalene) thereby re-dissolved. Smaller packages, e.g. drums, would likewise require gentle warming before being emptied. Barge-loads can be carried short distances and it is advantageous if the barge tanks are fitted with steam pipes. C.T.F. 50, being fluid down to 32° F., can be carried in any suitable container under any but the most severe weather conditions.

Hard pitch or Pulverisable Coal Tar Fuel is generally delivered in lumps ranging in size from 18 inches cube downwards. Before being fed to the pulveriser the lumps must be crushed into pieces  $\frac{1}{2}$ –1 inch in size, this operation being best performed by a spike roll crusher. If stored in the open the hard pitch should be kept in lump form as rain or casual water will then easily drain away. The pre-crushing operation should be delayed as long as possible and should immediately precede the pulverising process. The presence of water can increase the load on the pulveriser. The handling of the pulverised fuel is discussed later.

(iv) *General Properties of the Coal Tar Fuels.* As delivered, the coal tar fuels are technically free from water. They have a low sulphur content which never

exceeds 1 per cent. by weight and is generally considerably below that figure. They are almost ash-free, the amount of ash never exceeding 0.75 per cent. by weight, which is the maximum permitted in the solid pulverisable fuel. In the creosotes and the thinner tarry fuels it is almost negligible.

The gross calorific values of the fuels range from 17,000 B.Th.U. per lb. for C.T.F. 50 and C.T.F. 100, down to 16,000 B.Th.U. per lb. for C.T.F. 400 and the pulverisable fuel. The others have intermediate values. Calculated on a basis of 1 gallon of fuel at ordinary temperature the above-mentioned calorific values become 177,000 B.Th.U. and 200,000 B.Th.U. respectively.

Typical ultimate analyses of the fuels are given in Table 10.

TABLE 10. ULTIMATE ANALYSES OF COAL TAR FUELS

	Per cent.				
	C	H	S	N	O (by diff.)
C.T.F. 100 * .. ..	89.30	6.90	0.13	0.84	2.83
C.T.F. 100 † .. ..	86.50	8.00	0.25	0.84	4.41
C.T.F. 200 .. ..	89.86	6.05	0.39	1.11	2.52
C.T.F. 250 .. ..	89.97	5.90	0.44	1.16	2.45
C.T.F. 300 .. ..	90.08	5.73	0.49	1.22	2.39
C.T.F. 400 .. ..	90.42	5.23	0.65	1.38	2.19
Pulverisable C.T.F. ..	90.66	4.90	0.86	1.42	1.70

\* London horizontal retort tar oil.

† Midland vertical retort tar oil.

C.T.F. 50, being a thin mobile oil, can be pumped and atomised at ordinary temperature. When using C.T.F. 100 it is merely necessary to warm the fuel to slightly above the "fluidity-point" for it to be handled with equal facility. The temperatures to which the remaining liquid fuels must be heated for purposes of handling etc., are determined by their viscosity-temperature relationships. It has been explained that the title of each fuel indicates the approximate temperature in degrees Fahrenheit to which the fuel must be heated for efficient atomisation. It should not be necessary to exceed these temperatures at the burner nozzle.

For pumping, it is generally considered that the viscosity of the fuel should not exceed 12.5 stokes. A figure of 10 stokes will allow a reasonable margin, and for their viscosities to fall to this value the fuels must be heated to the temperatures indicated.

C.T.F. 200 .. ..	85° F.
C.T.F. 250 .. ..	140° F.
C.T.F. 300 .. ..	180° F.
C.T.F. 400 .. ..	270° F.

Coal Tar Fuels 200, 250, 300 and 400, among the liquid fuels, contain varying amounts of material insoluble in toluole. The maximum allowed ranges from 15 per cent. to 30 per cent. by weight, those being the figures for C.T.F. 200 and C.T.F. 400 respectively. This toluene insoluble material consists of resinous compounds of high molecular weight containing a high percentage of carbon. The particles are of microscopic dimensions and form a stable uniform dispersion in an oily medium.

These fuels should not be mixed with petroleum fuels except on expert advice. Contact of one with only small amounts of the other may cause local instability and coagulation with the possibility of a stoppage. It may be advisable to steam out the system when changing over from one fuel to the other.

(v) *Application of Coal Tar Fuels.* Considerable experience has now been gained in the use of coal tar fuels and with the aid of the information now available the design of new installations or the conversion of existing oil-burning units can follow well-established principles.

As a general rule, any fuel oil system which is fitted with tank heating and the usual flow heaters can comfortably handle either C.T.F. 50 or C.T.F. 100. When using C.T.F. 100 it may be necessary to increase the wattage of electrical heating equipment and, unless ring mains and gravity feed lines are drained

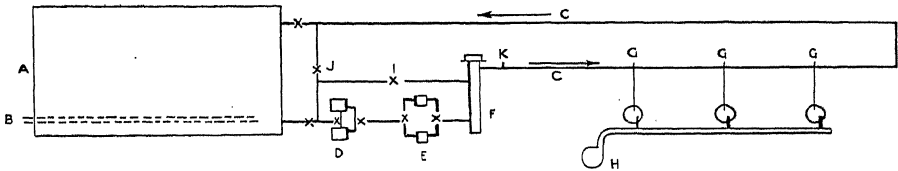


FIG. 9. General arrangement of installation for burning lighter tar oils.

- |  |   |
|--|---|
| A. Lagged storage tank.  | F. Thermostatically controlled flow heater raising fuel to atomising temperature. |
| B. Tank heating.   | G. "T" valves to low pressure air burners.  |
| C. Ring main containing steam tracer or electric heating cable within lagging. | H. Fan supplying atomising air.   |
| D. Duplicate steam jacketed filters with necessary "T" cocks.                  | I. Pressure relief valve.   |
| E. Duplicate pumping arrangement with necessary "T" cocks.                     | J. Ring main pressure regulating valve.   |
|  | K. Thermometer.   |

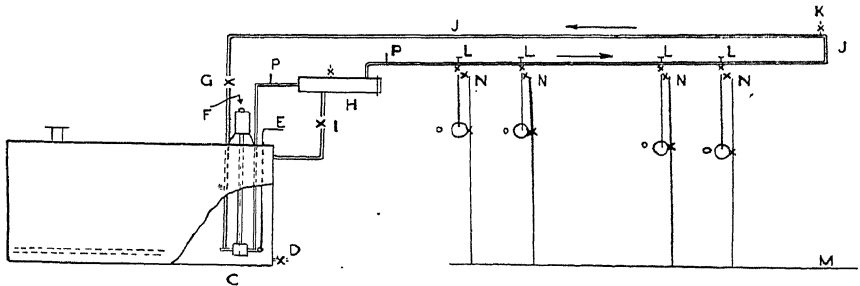


FIG. 10. General arrangement of installation for burning heavier tar oils.

- |  |  |
|--|--|
| A. Fuel storage tank.  | I. Combine flow heater drain cock and pressure relief valve.   |
| B. Steam heating coil or thermostatically controlled electrical heater.              | J. Ring main lagged with the inclusion of steam tracer line or electrical heating cables, graded downwards from vent cock "K." |
| C. Submerged gear pump.  | L. "T" type valves to steam jet burners O.   |
| D. Tank drain cock.  | M. Steam line.   |
| E. Pump drain cock.  | N. Steam valve for scavenging lines to burners.  |
| F. Pump motor.   | P.P. Thermometers.   |
| G. Ring main pressure balancing valve.   |  |
| H. Electrical flow heater thermostatically controlled at fuel atomising temperature. |  |

when closing down, such pipes should be provided with a steam tracer line or electrical heating cable. Pumps should also be kept warm.

The burning of the more viscous liquid fuels calls for a rather specialised technique. The best arrangement is to circulate the fuel, by pumping, from the storage tank through a ring main past the burners, and back to the tank. Gravity feed from the storage tank to the burners is quite practicable in small installations. The temperature of the fuel circulating in the ring main is limited by the pressure of the steam available. If any further boosting is required to raise the temperature of the fuel to the atomising point, electrical or other flow heaters can be installed in the system. At least  $2\frac{1}{2}$  times the fuel

consumption should be circulated through the ring main. This will stabilise temperature and pressure conditions in the system.

The storage tank and all pipe-work should be free from joints made with rubber asbestos compounds, and plain asbestos millboard is quite satisfactory. The fuel lines throughout should be steam-jacketed or lagged, with the inclusion in the lagging of a steam tracer line or electrical heating cable. Steam-jacketed pumps should also be used.

Typical installations are shown diagrammatically in Figs. 9 and 10.

The usual type of heavy oil burner can be adapted to most of the fuels. Pressure jet, controlled pressure jet, rotary, medium pressure air and low pressure air burners are all applicable to one or more of the less viscous fuels, such as the Coal Tar Fuels 50, 100, 200 and 250. For the more viscous grades, such as Coal Tar Fuels 300 and 400, medium pressure steam jet burners are generally employed, using steam at a pressure of 50–100 lb. per square inch. Suitable burners are offered by a number of manufacturers. Coal tar fuels are not suitable for use with very low capacity burners with correspondingly small orifices. Similarly, needle valves should be avoided and the ported type of valve employed.

Coal tar fuels have a slight corrosive action upon brass and similar non-ferrous metals. It is advisable therefore that valves, filters and other fittings should be made of iron or steel.

Pulverised coal tar fuel practice follows pulverised coal practice very closely, but the following differences may be noted:—

(a) The pulveriser should be of the air swept impact type and must be fed with cold air. Edge runner mills, ball mills, tube mills and others with a positive grinding action are not suitable.

(b) The velocity of flame propagation is of the order of 80 feet/second, and burner nozzle velocities should exceed this figure.

(c) Pipe lines conveying pulverised coal tar fuel should be suitably shielded and insulated from radiant heat.

(d) The burner nozzle must be a plain tube either air or water cooled. Spreader devices are not applicable.

(e) Ring main systems such as those used for pulverised coal are not suitable for pulverised coal tar fuel.

For conveying the pulverised fuel from the pulveriser to the burners, two methods are commonly practised. In the first, where the furnaces to be fired are of large capacity, the fuel is fed direct to the burners. In the second method, which is used when a number of furnaces are to be fired at irregular periods, the fuel is discharged into a pulverised fuel storage bin *via* a cyclone separator, and the fuel is then conveyed to the burners by independent fan and feeder units which may be started up individually or severally and which do not necessarily demand that the pulveriser be operating at the same time. The storage bins should be situated in cool, dry places.

The theoretical amount of air required to burn 1 lb. of coal tar fuel is of the order of 160 cubic feet, but in designing plant it is customary to base calculations on a figure of 200 cubic feet of air per lb.)

## ELECTRICITY

Electricity is used for furnace heating for special purposes to an increasing extent. Heat is converted into electricity at generating stations with a thermal efficiency which up to the present is never above 30 per cent. The average value for the generating station is about 20 per cent. It is then transmitted to the place of use and is reconverted into heat within the furnace. This conversion may be effected by one or more arcs, by primary or induced currents flowing through resistances, or by currents induced within the charge itself

(cf. Chapter XXI). Heating and annealing furnaces are usually of the resistance types ; steel melting furnaces operate through induced currents within the charge and with arcs ; arcs are used particularly when non-conducting material is to be heated to a very high temperature, for example in the manufacture of calcium carbide, and in the production of fused alumina.

Electricity is used in furnace work for special reasons rather than from considerations of heat energy cost though, where off-peak current can be supplied at a low price, cost considerations may be favourable, e.g. the use of resistance furnaces for carbonising coal in Canada. Electric furnaces are discussed in detail in Chapter XXI.

The loss of heat in the flue gases which is unavoidable with solid, liquid and gaseous fuels does not occur with electricity. Moreover, since the heat can generally be generated exactly where it is wanted, the higher efficiency in utilisation acts against the higher first cost per therm and may cause electrical furnaces to be more economical than other types under favourable circumstances.

## CHAPTER III

## THE STORAGE OF COAL

Oxidation of coal—Chemistry and physics of oxidation of coal—Practical principles of storage—Deterioration of coal in store (coking power, C.V., size, friability)—Watching temperatures—Dealing with fires—Storage of opencast coal.

THE object of storing coal is to provide reserves to keep plant working during a period when supplies may be dislocated. The dislocation may be caused by the interruption of transport facilities, due, for example, to the weather and the more normal delays incidental to rail transport. Furthermore, storing coal in the summer to meet the increase of consumption in winter

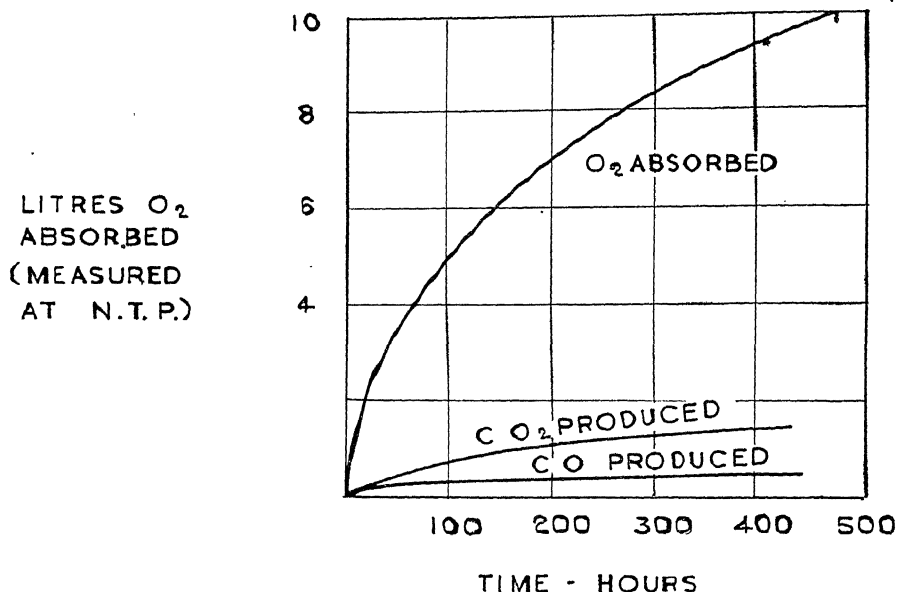


FIG. 11. Rate of absorption of oxygen by a coal.

(Reproduced by permission of the publishers (Longmans, Green) from "Coal: Its Constitution and Uses," by W. A. Bone and G. W. Hinshelwood.)

reduces the seasonal fluctuations in the demands on the mining and transport industries. Even under good conditions it is unlikely that coal will be delivered at exactly the rate at which it is consumed, and there must therefore be a stock that will absorb the difference from time to time existing between deliveries and consumption.

Stored coal, being exposed to the atmosphere, is subject to attack by the oxygen in the air. Under certain conditions the coal may oxidise so rapidly as to heat and ultimately ignite, and then the coal will depreciate greatly in value. Some coals merely disintegrate in storage. It is important to know how to store coal in order to minimise these unwelcome effects.

## THE OXIDATION OF EXPOSED COAL

Fresh coal absorbs oxygen when it is exposed to the air. Figure 11 shows typically how the oxygen absorbed by a coal passing a 90-mesh sieve exposed

in an atmosphere of oxygen at about 108° C. increases with the passage of time.

The absorption of oxygen is accompanied by the evolution of heat and there is therefore a rise in temperature.

At first the temperature rises only slightly, and if the heat produced can be dissipated the rise in temperature remains slight or even imperceptible and there are no serious consequences.

However, if the conditions do not favour the dispersal of the heat produced, the temperature will rise noticeably. As the temperature rises, the rate at which oxygen is absorbed also rises, and this in turn means an acceleration in the rate at which heat is evolved, so that the process is continually aggravated.

But as the temperature rises the rate at which the heat is dissipated increases. Eventually, a balance may be struck between the rate at which heat is produced and the rate at which it escapes, and then there will be no further rise in temperature, and in time, as the rate at which oxygen is absorbed slackens, the temperature will fall.

Finally, the heat evolved may not be dissipated fast enough to prevent the cumulative heating from leading ultimately to spontaneous combustion. The last stages of this heating are rapid, with the temperature rising at an increasing rate.

There is a critical temperature in the heating process ; when it is reached the temperature inevitably rises at an accelerating rate until combustion becomes active, and there is no possibility that the coal will cool and become safe before it fires. This critical temperature varies according to conditions, and in a heap of coal it is in or near the range 50° to 80° C. (122°–176° F.).

## THE CHEMISTRY OF OXIDATION

The absorption of oxygen is chiefly due to the reactive constituents in many kinds of coal. These compounds occur in greater proportion in coals of lower rank, that is in coals high in volatiles, than in the high rank coals which have a lower volatile content. The less mature or geologically more recent coals containing the least carbon and the most oxygen are the most easily oxidised.

Below about 80° C. (176° F.) oxygen is absorbed by coal and there is little if any formation of the gaseous oxides of carbon. At higher temperatures the reaction quickens, steam and oxides of carbon are evolved, and the process of combustion has started, although whether it can maintain itself depends on conditions. The temperature at which active combustion, visibly recognisable as such, occurs and progresses is much higher still.

As conditions are so variable, even for one variety of coal, there is no generally recognised or satisfactory method of measuring experimentally the liability of a coal to heat spontaneously. A method that may be found useful is to pass a stream of oxygen over the dried coal under prescribed conditions using various initial temperatures until a temperature is found from which the coal ignites in sixty minutes.\* There is experimental evidence that liability to spontaneous combustion can be linked with the results of this test somewhat as follows:—

Temperature from which the coal ignites in sixty minutes °C.	Degree of liability to spontaneous heating
100–109	dangerous
130–138	medium dangerous
above 150	not dangerous

\* Marian, described in Gluud, "International Handbook of the By-Product Industry," English edition, p. 78.

Figure 12 indicates the sequence of events and is based on the behaviour of fine coal in small quantities heated by external means in a current of air or oxygen ; these are conditions applied in laboratory investigations. The curve reproduces a complete history for such conditions, but needs to be interpreted for application to coal stored in heaps.

The self-sustained process of combustion, shown as starting at  $200^{\circ}$  to  $270^{\circ}$  C. ( $393$ – $518^{\circ}$  F.) does not start lower because the cooling effects operating under laboratory conditions are greater than the heating effects due to oxidation. In such circumstances the heating cannot become cumulative so as to be self-sustained.

In a heap of coal, where there is no external source of heat, the tendency for oxygen to be absorbed decreases as the affinity of the coal for oxygen is satisfied ; the rate at which oxygen is absorbed may therefore fall so that the rate at which heat is developed is less than the rate at which it is lost, when the

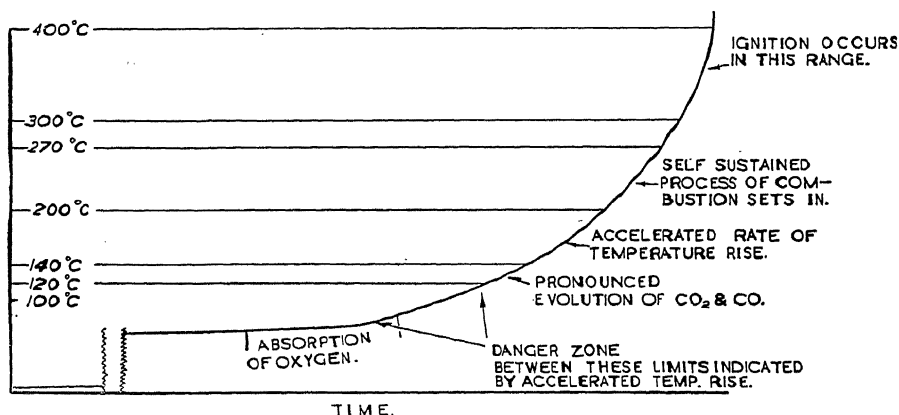


FIG. 12. Sequence of events in the oxidation and firing of coal, based on laboratory results.

curve will show a maximum, whereafter its slope is negative. This maximum usually occurs well below  $100^{\circ}$  C. ( $212^{\circ}$  F.), and the subsequent negatively sloping portion of the curve is indicated by a dotted line.

If the coal in a heap is not fine enough only the very earliest portion of the curve will be traversed, and the maximum with the subsequent negatively sloping portion of the curve will appear.

If the maximum does not appear, and the curve continues with a positive slope, heating is cumulative, and the whole of the curve may be embraced in the range of self-sustained combustion.

Before this can happen a predominant fraction of the coal must be finer than  $\frac{1}{8}$  or even  $\frac{1}{16}$  inch.

There will be some inverse relationship between the fineness of the coal and the scale of the time-axis.

## THE PHYSICAL CONDITIONS OF OXIDATION

It has been shown that there are in a store of coal two opposing processes : the production of heat due to oxidation and incipient combustion ; the loss of heat due to radiation, conduction and convection ; and that it is the balance between these two processes that decides the extent of heating.

Given access to oxygen, the production of heat varies with the surface exposed by the coal. For a given weight of coal the surface exposed is greater the finer



the coal ; and conversely it is less with larger coals, so that heat is produced more rapidly from finer coals. In fact, with coals in which no fines or dust occur the rate at which heat is produced is negligible and difficulties due to spontaneous heating are confined almost exclusively to coals in which fines or dust are admixed.

In the loss of heat the chief part is played by air passing through the heap and removing heat as sensible heat. Here again the temperature rise will be kept low in a heap of large coal because the texture of the heap is open and air-currents can pass through the heap readily ; whereas the greater production of heat when the coal is fine and the texture of the heap close, cannot be eased so readily because air-currents do not find a continuous passage through the heap.

It is because of this that ventilation of a heap of stored coal is practised. Experience suggests, however, that if a heap of stored coal would fire spontaneously it is difficult and expensive to ventilate it enough to make it safe, and that the danger can be more easily countered by suppressing ventilation, so that the amount of air entering the stack is insufficient to support appreciable oxidation and heating.

### THE PRACTICAL PRINCIPLES OF COAL STORAGE

Some of the practical principles to be observed in safely storing coal can be deduced from what has already been said, and these principles can be summarised as follows :—

	Conditions favourable to safe storage	Conditions unfavourable to safe storage
Size of coal	Nuts, cobbles, round coal ; exclusion of fine coal.	Fines, slack, run-of-mine, alone or included with large coal ; smalls formed by breakage, such as falling from a chute.
Size of store	Less than about 200 tons.	More than 200 tons.
Height of store	Not greater than 8 to 10 feet.	Greater than 10 to 12 feet.
Ventilation	Adequately promoted ; or what is generally easier, effectively suppressed.	Moderate or natural, especially if due to passage caused by segregation during stacking, owing to a pocket of large spaces against pillars or walls.
Age of coal	Greater than 3 to 6 months after mining.	Up to 10 weeks.
State of coal	Having heated and cooled.	Freshly wrought.
Type of coal	Hard or dull, anthracite, anthracitic coal, splint, cannel.	Soft and bright ; bituminous, sub-bituminous, lignite.
Site of stack	Most open situations on clean, firm ground.	Near a source of heat, such as a boiler house wall, or overlying steam pipes, or mixed with organic waste or oily material.

While it is advocated that ventilation should be suppressed, in a small heap or one not too high the ventilation that does occur and the distribution of air currents will help to keep conditions safe, and with the smallest heaps the suppression of ventilation will not be so urgent ; the ventilation may even be promoted.

The liability to oxidation is lessened with increasing age of the coal as some of the unstable constituents become oxidised. Old coal will not therefore be as susceptible as freshly wrought coal.

### THE SIZE OF THE COAL AND THE CORRECT METHOD OF STACKING

Small coal is the cause of nearly all storage difficulties. In coals of mixed sizes, segregation is the most frequent cause of spontaneous heating, because the larger pieces of coal open the texture of the heap and facilitate the type of

ventilation necessary for heating to proceed. By coals of mixed sizes are understood coals that include fines, and run-of-mine coal. Uniformly fine coal is, if anything, less liable to heat than mixed sizes.

Singles are likely to be safe and larger sizes are quite safe, provided that fines have not been produced during handling. The danger of heating is greater under a chute where the finest coal falls or is formed by breakage. Sized coal should not therefore be dropped from a height when it is being stacked. Stacking coal by a drag-line conveyor has the advantage of producing less fines than dropping it from an overhead conveyor, and is thus less likely to lead to firing.

The coal should be spread out as it is stacked so that there is no local segregation of sizes, and lumps should be distributed as uniformly as possible among the fines.

### THE SUPPRESSION OF VENTILATION

Ventilation, especially in a large heap of coal, can be suppressed to a degree that is effective more easily than it can be promoted. This is because attempts to ventilate large heaps often do not go far enough, although ventilation is practised by some users with considerable success (cf. Dunningham and Grumell, *J. Inst. Fuel*, 1937, 10, 170).

Ventilation can be reduced by avoiding segregation and packing the coal tightly and uniformly and by blanketing the heap with fine coal. If the coal can be kept damp, ventilation is reduced still further because moisture decreases or even fills the spaces between particles, impeding the movement of air.

The stack may be walled to reduce ventilation. A rough wall of the larger rectangular lumps of coal is built. A permanent coal storage site may have a low wall most of the way round it to protect the base of the stack whence ventilating currents start.

The wind assists ventilation, and to reduce its effect the stack is best built elongated with its longer sides running in the direction of the prevailing wind; this generally means south-west to north-east.

A method that is completely effective when the all too rare facilities are available is to store coal under water, for example in a bunker that can be flooded; or to store it in an atmosphere of flue gas.

### THE DIMENSIONS OF THE STACK

Trouble seems rarely, if ever, to arise in heaps of less than 200 tons. It is usually easier to control the height of the stack than to limit the weight of coal in it. A safe height seems to be about 8 feet, but if the coal is of a suitable size and type it may go up to 16 feet. Hard nuts have been stored safely up to 20 feet and larger coals and anthracite up to 30 feet.

A stack may be built up in stages, whereby the lower layers have had an opportunity to pass over their critical stages before the upper layers are added. Thus, a 6-foot stack may be increased to 10 feet after six months, or by 18 inches every three months; or an 8-foot stack may be built up to 16 feet after nine months.

The advisability of adding to the heap may be judged by the temperatures in the existing part.

The height of the stack should follow the contour of the ground.

Coal occupies from 40 to 50 cubic feet per ton; the more mixed sizes give the more compact heaps, and the heap is also more compact when the coal is packed carefully.

### THE SITE OF THE STACK

A concrete floor provides an ideal base for a stack, but a well rolled floor of ashes may be satisfactory; hard clay or chalk may also be satisfactory. Peaty

ground should be dressed with a few inches of ballast or ashes and then rolled to make it as firm as possible.

An important consideration in the preparation of the floor is the clean and complete recovery of the coal. Apart from this, vegetable matter and waste organic matter should always be cleared as it may contribute to the start of heating.

A wet site is not necessarily advantageous. Storage under cover offers no advantage to compensate for the outlay in providing the cover.

### THE PERIOD OF ACTIVITY IN A COAL STORE

Coal oxidises most rapidly when it is freshly mined, and the rate at which it is affected decreases as time passes.

Spontaneous combustion usually occurs within four weeks to four months after stacking. A coal that is eight months old can be regarded as safe, if there is no heating in the heap, or if what heating there is is not increasing.

It has been reported by an undertaking that stores large quantities of coal that there is considerable risk in stacking freshly wrought coal in contact with coal which has been in stock for some time. Experience has shown that this point of junction is a fruitful site for fires.

### THE DETERIORATION OF COAL ON STORAGE

Deterioration obviously depends on the amount of heating that goes on in a store of coal, but if there is no appreciable heating, deterioration is at a minimum and although data are not available to indicate the depreciation of the various qualities of coal with time, it is possible and useful to know the order of the amount of deterioration to be expected.

Bright coals—clarain and vitrain—are affected more than dull and hard coals—durain. A freshly broken surface is the most active for the absorption of oxygen and the surface of old coal becomes dull. This is said to affect the market value of the coal, but alone it does not indicate any noticeable depreciation in quality.

### THE LOSS OF COKING POWER

Of all the properties of a coal, the coking power is easily the most rapidly and extensively affected. Frequently the effect is very marked when there is no other indication of a change.

The coking power falls off much more rapidly during the first few months of storage than later. It is not certain that a coal ever loses all its coking power when it is stored, but coals that are originally weakly coking may have their coking power reduced to insignificant proportions. The rate at which coking power is reduced is also seriously affected by heating.

If coking power is an important characteristic of the coal to be stored, a coal should be selected with a higher coking power than would otherwise be required, so that the excess of its coking power acts as a reserve.

### THE LOSS IN CALORIFIC VALUE

Steam-raising power is often diminished by exposure to the weather and atmosphere, and is not unnaturally ascribed to a loss in calorific value. If the coal has not heated appreciably the loss in calorific value is slight, so slight as a rule as to be imperceptible. The loss in steam-raising power is likely to be due rather to an increased loss of small coal as unburnt matter through the grate.

If coal that is stored does not heat appreciably, the loss in calorific value rarely exceeds 0.5 to 1 per cent. per annum. An examination of twenty Scottish gas coals (Jamieson and Skilling, "The Deterioration of Coal on Storage," *Trans. Inst. Gas E.*, 1938, 88, 663) showed a mean loss of calorific value of 0.7 per cent. in upwards of  $3\frac{1}{2}$  years.

If the coal heats, the loss in calorific value may reach any reasonable value, according to the extent of heating.

### THE EFFECT OF WEATHERING ON THE SIZE OF STORED COAL

Coal may disintegrate under the action of the weather and small or fine coal may thus be formed. This formation of fine coal will not greatly affect the tendency of the coal to heat as the coal is always likely to oxidise faster than it disintegrates.

Examples of the extent of disintegration are afforded by work done on Scottish coals (Jamieson and Skilling, *loc. cit.*) and by the Fuel Research Board (Annual Reports, 1937, 69; 1938, 83; 1939, 60). Fourteen Scottish gas coals from which all coal passing a  $1\frac{1}{2}$  inch screen was removed contained from 3 to 20 per cent. of coal passing a  $1\frac{1}{2}$  inch screen after 2 years of exposure and from  $\frac{1}{4}$  to 28 per cent. after 44 months. Six similar coals had originally from 30 to 45 per cent. of coal passing a 1-inch screen, and this size was increased by percentages varying from zero up to twelve. At the Fuel Research Station, midland, north-country and Welsh gas, house and steam coals were stored for periods up to a year. The results are summarised in Table 11.

TABLE 11

Period of storage—months	0	6	9	12
S. Wales steam, $2\frac{1}{2}$ – $1\frac{3}{8}$ in., per cent. .. ..	100	93.2	91.6	90.7
Warwickshire house, $2\frac{1}{2}$ – $1\frac{3}{8}$ in., per cent. .. ..	100	91.4	90.0	87.3
Nottinghamshire house and steam, $2\frac{1}{2}$ – $1\frac{3}{8}$ in., per cent. .. ..	100	82.2	73.1	69.4
S. Yorkshire gas R, above 2 in., per cent. .. ..	43.7	37.4	34.7	30.8
S. Yorkshire gas T, above 2 in., per cent. .. ..	54.0	46.3	48.2	47.8

Coal does not seem to break down progressively to the smallest sizes. The process appears to cease eventually when "natural grades" have been produced (Scarff, Hall and Yearsley, "The Storage of Coal: the Effects of Storage without Rise in Temperature," *Trans. Inst. Min. Eng.*, 1941, Vol. C, pt. 4, 88–129) owing to the splitting of the coal along bedding planes and at partings (Annual Reports of the Fuel Research Board, *loc. cit.*).

### THE FRIABILITY OF WEATHERED COAL

The Warwickshire and Nottinghamshire coals used in the investigations at the Fuel Research Station (*loc. cit.*, 1938, 83) were subjected to a tumbler type of test in a rotating drum in which the coal was dropped a few inches a few times in each revolution. Some of the results of these tests are reproduced in Table 12. These results show that a coal when it is handled after weathering will break more easily. This is important in boiler firing and other furnace work.

With small coal disintegration does not matter, but with large sized coal it

TABLE 12

Period of storage—months	0	6	9	12
Warwickshire house, $2\frac{1}{2}$ – $1\frac{3}{8}$ in., per cent. .. ..	87.7	75.9	74.6	68.1
" " $1\frac{3}{8}$ – $\frac{3}{4}$ in., " " .. ..	6.9	13.3	14.8	16.7
Nottinghamshire house and steam, $2\frac{1}{2}$ – $1\frac{3}{8}$ in., per cent. .. ..	81.4	58.0	49.1	49.7
" " " " $1\frac{3}{8}$ – $1\frac{1}{4}$ in., " " .. ..	9.6	15.1	16.8	17.1
" " " " $1\frac{1}{4}$ – $\frac{3}{4}$ in., " " .. ..	4.7	12.1	14.4	13.7

may be important. The size may be an essential quality of the coal, and in furnace work for example it may mean that there will be an undue loss of combustible material through the grate, and this, as previously indicated, will produce the impression that there has been a decrease in the calorific value.

Cannels and anthracites disintegrate little if at all, and hard, dull coals disintegrate less than soft, bright ones.

## THE DURATION OF STORAGE

As deterioration is most rapid in the earlier months of storage and its rate is always decreasing—provided heating does not eventually accelerate it—stored coal should not be used while fresh supplies are available. In this way the loss is least because a new lot of fresh coal is not being exposed continually, and the rate at which the stored coal deteriorates is always falling; moreover, as the second handling of the stored coal is deferred, labour costs are reduced.

## PRECAUTIONS TO BE TAKEN IN A STORE OF COAL

At regular intervals of about a week during the first three months of storage, the temperatures of different places in the stack should be observed.

A rough method that may be found suitable for doing this consists in inserting metal rods into the heap at intervals of 10 to 20 feet, with their ends projecting, and every few days feeling how hot these projecting ends are.

It is better, however, to use wrought-iron or steel pipes closed at the lower end and to take temperatures by lowering a maximum-reading thermometer down them. The pipes should be built into the stack as it is made up. They should be from 15 feet to 50 yards apart according to the size of the stack, the kind of coal and economic considerations. The temperatures can then be read at various depths and should be recorded on a chart. Once the danger period is past the readings can be discontinued. Spontaneous combustion is generally considered most likely to occur within four weeks to four months after stacking.

The critical temperature varies considerably according to conditions and is generally taken to lie between 55° and 70° C. (131° and 168° F.), but it may be lower if conditions are unfavourable. It can only be determined to within about 5° C. (9° F.) by laboratory experiments.

A slow and steady rise in temperature must be expected, when the critical temperature is reached, to accelerate rapidly and culminate in a fire. Adequate graphing or logging of the temperature-history of the respective points in the stack will give clear warning of this.

## DEALING WITH FIRE IN A COAL STACK

When the critical temperature is reached, it is wrong to delay in the hope of a spontaneous improvement in conditions. If the precautions are not taken forthwith, a fire is inevitable, and the affected area will spread.

The danger usually occurs at points from 3 to 7 feet below the surface, where ventilation is sufficient to promote heating and the depth of coal sufficient to prevent a rapid enough loss of heat. Temperature observations will indicate the exact zone, which may also be shown by a light grey smoke and a distinctive odour, though by this time the fire will be established.

The seat of the fire must be dug out by digging a hole, trench or valley, completely exposing the hot place, and scattering or using the hot coal.

A crane and grab or similar mechanical appliance greatly simplifies this operation. If a fire has started, spraying with water is ineffective unless a very large volume can be used.

In the early history of a stack of coal, however, spraying with water has a twofold advantage. It interferes with ventilation and may restrain the tem-

perature below its critical level while the heap is piloted through its peak heating period.

### THE STORAGE OF OPENCAST COAL

A recent development in coal-getting is a reversion to the distant past in the opencast working of outcrop coal. Under suitable conditions a seam, as it dips from its bassett edge, is worked with suitable types of excavating machinery. The output of a productive seam becomes large at an early stage in the development of operations, and the storage of some or all of the coal close to the working or at selected storage sites becomes necessary or may form part of the policy of disposing of it.

Generally, the bassett edge of the seam is very inferior and has to be discarded. As the seam is followed in, the coal may remain rather poor, but nearly always improves rapidly, at times approaching pit-coal in quality; it is usually weathered to various degrees and therefore generally has a high oxygen content, with a corresponding reduction in its carbon and hydrogen contents and its calorific value. The oxidation entails a high inherent moisture content and generally a high degree of friability. On the other hand, it is not uncommon for opencast coal to have a lower ash content than the coal from the same seam from a nearby pit; and opencast coals with an unexpectedly high coking power have been won.

Table 13 gives a few examples of opencast coals. It must be remembered that the coals of this country, as normally mined, cover a wide range of types. When these coals are worked at their outcrops, further differences, varying rapidly with the depth of cover and with other conditions, are imposed on the original differences. The figures given in Table 13 therefore serve only as examples and are not intended to be representative.

TABLE 13

	Nottinghamshire			South York- shire	South Wales
	(a)	(b)	(c)		
Moisture as received, per cent. . . .	31.0	15.9	—	8	8.4
Air-dry basis					
Moisture, per cent. . . . .	10.3	7.8	3.4	3.2	4.0
Volatile matter, less moisture, per cent.	33.4	32.1	33.6	33.6	29.5
Fixed carbon, per cent. . . . .	50.8	58.8	59.0	60.7	61.5
Ash, per cent. . . . .	6.3	1.2	4.0	2.5	5.0
Dry, ash-free basis					
Volatile matter, per cent. . . . .	40.0	35.3	36.4	35.6	32.4
Calorific value, B.Th.U./lb. . . . .	11,200	13,120	14,700	14,780	14,630
Carbon, per cent. . . . .	70.7	77.0	82.3	—	—
Hydrogen, per cent. . . . .	3.8	4.8	5.3	—	—
Sulphur, per cent. . . . .	—	—	—	0.78	0.75

The Nottinghamshire examples are from (a) near the bassett edge, (b) from further in, and (c) for purposes of comparison, from the same seam as won in a nearby pit. The South Yorkshire and South Wales coals are examples of good opencast coals, though these are not the best that can be got.

The highly oxidised opencast coal may not be liable to serious spontaneous heating. But generally it would be unwise to assume that the coal is any less liable than pit coal. The coal is in some ways of a new type, and is often stored under conditions much less favourable to the avoidance of trouble than in

ordinary times. Until experience shows that in any particular set of circumstances the normal precautions for storing coal are unnecessary, it is not advisable to relax any of the precautions that would otherwise be practised. This is especially so as the quality of the coal improves steadily as the working of the seam progresses.

Occasionally at sites for the storing of opencast coal, space may not be seriously limited, and then there is no difficulty in confining the height of the stack to 8 feet, which is a compromise between a height low enough to be quite safe and an economic use of the area of the site.

Unfortunately, however, ground is usually more precious and then it is necessary to bestow every precaution possible on the formation of the heap.

The whole of the site on which the coal is stored should be used at once so that the coal is built up all over the heap in thin layers. Owing to the great range and uneven mixture of sizes in opencast coal, it is very easy to get a segregation of large coal at the bottom of the stack so that air passes freely through large channels traversing the whole of the base of the stack; and this condition is made worse by occasional vertical or oblique veins of large coal running from top to bottom through the heap. No effort should be spared to avoid this very dangerous segregation.

If excavating machinery of a kind that can dig into the heap is available, and in connection with opencast working it often is, precautions need not be so stringent when these appliances can be brought to bear on the heap whenever it is necessary. If, when trouble occurs, the heap has to be dug into by hand, everything that can be done to ease this operation is helpful. In particular, if the coal is disposed in long narrow heaps, a heated portion can be isolated by a valley cut across the heap. A long narrow heap is also sometimes easier for certain kinds of excavating machinery to deal with than a compact one.

There should be no attempt at ventilation, unless on the very smallest heaps—say of 200 or 300 tons. If the coal can be compacted in its heap by the traffic of lorries and dumping cars over it, much will be done to impede the percolation of air through the coal.

At appropriate points in the heap, where they will not interfere with traffic handling the coal, pipes or rods can be inserted for observations on the temperature of the heap. A temperature of 50° C. (122° F.), especially if the rise in the temperature is occurring at an accelerating rate, indicates dangerous conditions. A dangerous situation must never be expected to improve spontaneously.

When a fire, according to observations of the temperature in the stack or other signs, appears imminent, the heated coal must be dug out and used or scattered in a thin layer over adjacent parts of the heap or on neighbouring ground.

If the coal shows signs of heating the condition may be satisfactorily reduced by steadily spraying water over the affected area in the form of a strong rain for some days. If the heating is more serious and is localised, the area may be dealt with effectively by running water on to it from a hose. It should be realised, however, that water is not necessarily a reliable way of meeting such a situation. This is especially true if a diffused heated area is treated with a jet of water that only covers a small area, for a strong flow of water in one place will find a preferred path down through the heap instead of percolating all through the area that needs treatment.

Spontaneous heating may become apparent within six weeks, but is not likely to show after about four months. There have been indications that if opencast coal has heated and the temperature has been reduced by suitable treatment it is not likely to heat again.

The general nature of stored opencast coal, both as to quality and size, makes it likely that deterioration will be negligible.

## CHAPTER IV

# THE CHEMICAL PRINCIPLES OF COMBUSTION AND GASIFICATION

Combustion and gasification—Chemistry of combustion—Calculation of analysis of flue gas from solid and gaseous fuels and of air requirements—Chemistry of gasification.

## COMBUSTION AND GASIFICATION

THE art of *combustion* consists in causing chemical reactions which generate heat to take place between the combustible material (generally carbon, hydrogen or compounds of these elements) and oxygen (generally in the form of air) in such a way that the maximum amount of heat is released. A secondary, but none the less important, branch of the art is to secure the type of flame needed for any particular industrial operation, e.g. an intensely hot flame or a slow, "lazy" flame of lower temperature.

Combustion of coal is brought about by the combination of the carbon (C) and hydrogen ( $H_2$ ) with the oxygen ( $O_2$ ) contained in the air. When carbon burns completely, it results in the formation of a gas, carbon dioxide ( $CO_2$ ). When carbon burns incompletely it forms carbon monoxide (CO) which when compared with carbon dioxide contains only half the amount of oxygen per unit weight of carbon. The combination of hydrogen with oxygen forms water vapour ( $H_2O$ ).

Air may be considered for combustion purposes as a mixture of 79.1 per cent. of nitrogen with 20.9 per cent. of oxygen by volume. As far as burning is concerned, the oxygen is the only useful part. The nitrogen is an adulterant ; it does not play any part in the combustion and it has to be heated up to the temperature of the other gases by the expenditure of part of the heat generated in the process.

The object of *gasification* is to cause chemical reactions to take place between carbon, oxygen and steam that will give rise to a combustible gas of the highest practicable calorific value.

Whereas the art of combustion is directed to the production of heat, the art of gasification is directed to the conversion of the solid into a gaseous fuel which is ultimately burnt to produce heat in another part of the plant.

Gasification is used where solid fuel is unsuitable or inconvenient for the process for which heat is required.

## COMBUSTION

Fuels consist essentially of carbon, hydrogen and oxygen combined to form more or less complex bodies ; the hydrogen may be uncombined and it is present as free hydrogen in most gaseous fuels ; similarly the carbon may exist uncombined as in coke. In combustion the compounds and elements are alike burnt to carbon dioxide and water vapour. In this process heat is developed. Combustion technique is thus concerned primarily with the development of the maximum quantity of heat from the fuel ; it must also be concerned with the quantity of heat that escapes unused from the plant. It is obviously necessary to understand the principles involved and to have means for measuring quantities of heat.

In this chapter the chemistry of combustion and gasification is described, leaving the thermal aspects of the subject to Chapter V.

## THE CHEMISTRY OF COMBUSTION

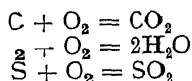
The products of complete combustion are carbon dioxide and water when the



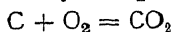
substance burnt is coal, coke, oil, gas or other fuel in normal use which contains as combustibles carbon and hydrogen. Since the combustion is effected with air the carbon dioxide and water are accompanied by a large amount of nitrogen and any excess oxygen that may have been used. The products of combustion are called "flue gas" or "waste gas."

Because most fuels contain small quantities of sulphur the flue gas also comprises oxides of sulphur, mainly the dioxide ( $\text{SO}_2$ ), but with some trioxide ( $\text{SO}_3$ ) in addition.

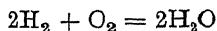
In the simplest chemical language these changes are expressed by the equations:—



Such equations not only indicate what substances take part in the reaction, but also the proportions in which they take part. Thus the equation,

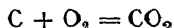


means that 1 atom of carbon combines with 2 atoms of oxygen to form 1 molecule of carbon dioxide which therefore contains 3 atoms—1 of carbon and 2 of oxygen. Similarly,



means that  $2 \times 2 = 4$  atoms of hydrogen combine with 2 atoms of oxygen to give 2 molecules of water. The number of atoms of the several elements on each side of the equation must be equal, since matter is indestructible.

Since the atom and molecule of any substance possess definite weights and the relative weights of the atoms of the several elements are known, it follows that these equations show also the weights in which the various substances take part. For example, since the atomic weight of carbon is 12 and that of oxygen is 16,



also indicates that 12 parts by weight of carbon combine with  $2 \times 16 = 32$  parts of oxygen to give  $12 + 2 \times 16 = 44$  parts of carbon dioxide. Similarly, since the atomic weight of hydrogen is unity,



implies that  $2 \times 2 = 4$  parts by weight of hydrogen combine with  $2 \times 16 = 32$  parts of oxygen to give  $2 \times (2 + 16) = 36$  parts by weight of water.

"Parts by weight" may be grams, kilograms, pounds, tons and so forth as desired provided the same units are used throughout.

In reactions involving gases, simple relations by volume also exist between the gases taking part because the molecular weight of gases expressed in the same units of weight occupy the same volume.

The molecular weights (expressed in the nearest whole numbers) of some of the more common gases are:—

hydrogen	..	2	nitrogen	..	28
oxygen	..	32	water vapour	..	18
carbon dioxide	..	44	sulphur dioxide	..	64

These weights of gases, dry and expressed in grams, occupy 22.412 litres at  $0^\circ \text{C}$ . ( $32^\circ \text{F}$ .) and 760 mm. bar. (30 inches bar.)\* If the weights on the

\* Although 760 mm. is 29.92 inches, it is convenient and sufficiently accurate for most practical purposes to regard 760 mm. bar. as equivalent to 30 inches bar.,  $60^\circ \text{F}$ . and 30 inches bar. being taken as the standard temperature and pressure for gas measurement in English units and  $15^\circ \text{C}$ . (or  $0^\circ \text{C}$ .) and 760 mm. for metric units.

basis are expressed in pounds, these several weights of gases all occupy 359 cubic feet, again at 0° C. (32° F.) and 760 mm. bar. (30 inches bar.). The volume occupied at any other temperature and pressure may be calculated from these data, the volume being directly proportional to the absolute temperature and inversely proportional to the pressure.

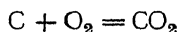
If a volume of gas  $V$  is measured at  $t_1$ ° C. and  $p_1$  mm. mercury pressure, its volume at  $t_2$ ° C. and  $p_2$  mm. pressure is

$$V \cdot \frac{t_2 + 273}{t_1 + 273} \cdot \frac{p_1}{p_2}$$

Similarly if the volume is measured at  $t_1$ ° F. and  $p_1$  inches bar., its volume at  $t_2$ ° F. and  $p_2$  inches bar. is

$$V \cdot \frac{t_2 + 460}{t_1 + 460} \cdot \frac{p_1}{p_2}$$

In the equation :



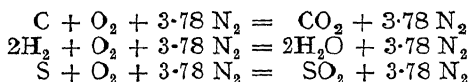
1 volume of oxygen gives rise to 1 volume of  $CO_2$ . For the reaction



2 volumes of hydrogen and 1 volume of oxygen combine to form 2 volumes of steam.

A "volume" may be taken as any desired quantity, such as 1 cubic foot, 1,000 cubic feet, 1 litre, 1 cubic metre, etc., provided that the same units are used throughout.

Since air may be taken to consist for practical purposes of 20.9 per cent. by volume of oxygen ( $O_2$ ) and 79.1 per cent. by volume of nitrogen ( $N_2$ ), 1 volume of oxygen in air is accompanied by (79.1/20.9 =) 3.78 volumes of nitrogen, and these equations from the point of view of the combustion engineer are more correctly written :



From these chemical relationships it is possible to calculate the composition of the flue gases from a coal or coke or oil. To assist in making the calculation, Table 14 has been derived from the simple chemical equations just discussed, it being assumed throughout that the steam ( $H_2O$ ) produced from combustion of the hydrogen remains uncondensed as a constituent of the gases.

TABLE 14

1 lb. of	Requires lb.			Yields lb.				Yields cu. ft. Measured at 60° F. and 30" bar.			
	$O_2$	$N_2$	Air	$CO_2$	$H_2O$	$SO_2$	$N_2$	$CO_2$	$H_2O$	$SO_2$	$N_2$
C to $CO_2$	2.667	8.833	11.500	3.667	—	—	8.833	31.4	—	—	119
$H_2$ to $H_2O$	7.939	26.287	34.226	—	8.939	—	26.287	—	188	—	353
S to $SO_2$	1.000	3.305	4.305	—	—	2.0	3.305	—	—	11.5	44

Density—lb./cu. ft. at 60° F. and 30" dry

Specific volume—cu. ft. at 60° F. and 30" per lb.

$CO_2$	0.117	$SO_2$	0.173	Air	0.0476	$CO_2$	13.43	$SO_2$	5.78	$H_2O$	21.00
$N_2$	0.0744	$O_2$	0.0846	Air	0.0765	$O_2$	11.82	Air	13.07		

The oxygen in the coal is combined with carbon, hydrogen and other elements. It is assumed for the purpose of calculation that this oxygen is combined wholly with hydrogen. An allowance of hydrogen equivalent to the oxygen present must be deducted from the effective weight of the hydrogen. This deduction is equal to one-eighth of the oxygen content, since 1 lb. of hydrogen combines with 8 lb. of oxygen to form 9 lb. of water.

### CALCULATION OF THE COMPOSITION OF FLUE GASES FROM COAL

To show how Table 14 may be used, and as the simplest method of illustrating the procedure, the composition of the flue gas (assuming that no excess air is used) is here calculated from a coal having the following composition :—

	Per cent
Carbon .. ..	74
Hydrogen .. ..	5
Oxygen .. ..	5
Nitrogen .. ..	1
Sulphur .. ..	1
Moisture .. ..	9
Ash .. ..	5

100

Water consists of 1 part of hydrogen to 8 parts of oxygen by weight. The first step is to assume that the 5 per cent. of oxygen in the coal is combined with one-eighth of its weight of hydrogen to form water as just indicated.

The water formed in this way is derived from the hydrogen that has combined with this oxygen (i.e.  $\frac{5}{8}$  per cent. of the coal) plus the oxygen contained in the coal (5 per cent.). With 9 per cent. of water already existing in the coal, the water liberated as steam is :

$$9 + 5 + \frac{5}{8} = 14.6 \text{ per cent. of the coal by weight.}$$

The remaining hydrogen (the "available hydrogen" as it is termed) is thus  $5 - \frac{5}{8} = 4.4$  per cent.

This 4.4 per cent. of available hydrogen will require oxygen from the air for its combustion, and will yield a further quantity of steam.

Thus from 1 lb. of coal there is derived :—

Incombustibles :—

	Per cent.	lb. per lb. of coal
Nitrogen ..	1	0.01
Ash ..	5	0.05
Water ..	14.6	0.146

Combustibles requiring air for their combustion :—

Carbon .. ..	74	0.74
Hydrogen .. ..		0.044
Sulphur .. ..		0.01

The next step is to consider the quantity of each element present in 1 lb. of the coal and to determine the quantity in cubic feet of the gases to which it gives rise, using the factors given in the preceding table. Due account must be taken of the nitrogen which accompanies the oxygen in the air required for combustion, and this is done in Table 14. The calculation, using the data given in Table 14, is as follows :—

# CHEMICAL PRINCIPLES OF COMBUSTION AND GASIFICATION

Weight of constituent		Volume of products of combustion— cu. ft.			
	lb.	CO <sub>2</sub>	H <sub>2</sub> O	SO <sub>2</sub>	N <sub>2</sub>
C	0.74	23.3	—	—	88
H	0.044	—	8.27	—	15.5
S	0.01	—	—	0.11	0.44
H <sub>2</sub> O	0.146	—	3.06	—	—
N	0.01	—	—	—	0.13
Ash	0.05	—	—	—	—
	1.000	23.3	11.33	0.11	104.07

At higher temperatures all these four gases are contained in the products of combustion ; if the gases are cooled to atmospheric temperature, as they would be if they were to be analysed by gas analysis apparatus, the bulk of the water vapour would condense. The following figures represent the difference between the gases in the flues and in the gas analysis apparatus :—

Total wet gases if calculated to 60° F. 138.81 cubic feet per lb. coal  
 „ dry „ „ „ „ 127.48 cubic feet per lb. coal

Composition by volume of flue gas (the coal being burnt with the exact theoretical amount of air) is obtained, for example, thus :—

$$\text{CO}_2 \text{ on wet gas} = \frac{23.3}{138.81} \times 100 = 16.7 \text{ per cent. by volume}$$

and

$$\text{CO}_2 \text{ on dry gas} = \frac{23.3}{127.48} \times 100 = 18.3 \text{ per cent. by volume.}$$

The complete analysis is :—

	On wet gas Per cent.	On dry gas Per cent.
CO <sub>2</sub>	16.7	18.3
SO <sub>2</sub>	0.1	0.1
H <sub>2</sub> O	8.2	—
N <sub>2</sub>	75.0	81.6

If excess of air is used, its volume must be added to the total as here calculated. The nitrogen in the air supply was 104.07 — 0.13 = 103.94 cubic feet per lb. of coal. If 50 per cent. of excess air is used in the example given, the air used for combustion is :—

103.94 × 100/79.1, or 131.4 cubic feet (= N<sub>2</sub> expressed as air)  
 Add to this 50 per cent. of 131.4, i.e. 65.7 cubic feet (= excess air)

Total air used per lb. of coal =            cubic feet

Total waste gases (wet) :

$$(\text{dry}) = 127.48 + 65.7 = 193.18$$

The 65.7 cubic feet of excess air comprises 52 cubic feet of nitrogen and 13.7 cubic feet of oxygen.

The complete analysis is then :—

	Cu. ft. from 1 lb. coal		Composition	
	Wet gases	Dry gases	Wet Per cent.	Dry Per cent.
CO <sub>2</sub>	23.3	23.3		12.0
SO <sub>2</sub>	0.11	0.11	0.1	0.1
	11.33	—	5.6	—
	156.07	156.07	76.2	80.8
	13.7	13.7		7.1
	204.51	193.18	100.0	100.0

### COMBUSTION DATA FOR FUELS

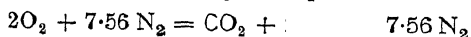
Proceeding in the manner indicated for a number of fuels, Table 15 can be derived on the assumption (already made in the preceding calculation) that all the fuels are burnt with the exact amount of air required for combustion.

TABLE 15

	Theoretical amount of air required per lb. fuel		Vol. of total waste gases cu. ft. at 60° F. and 30" bar.	Calculated wet waste gas analysis			CO <sub>2</sub> as determined by analysis i.e. with the water in the gas condensed
	lb.	cu. ft. at 60° F. and 30" bar.		CO <sub>2</sub> Per cent.	N <sub>2</sub> Per cent.	H <sub>2</sub> O Per cent.	
Non-caking coal 40% V.M. ; 5% ash; 10% moisture ..	8.68	113	122	17.2	73.8	9.0	18.9
Coking coal 30% V.M. ; 5% ash; 2% moisture ..	10.75	140	147	17.3	75.8	6.9	18.6
Low V.M. coal 18% V.M. ; 4% ash; 1% moisture ..	11.28	148	152	18.0	76.6	5.4	19.0
Anthracite 4% V.M. ; 3% ash; 1% moisture ..	11.34	150	152	18.9	77.4	3.7	19.6
Coke 1% V.M. ; 7% ash; 2% moisture ..	10.22	134	135	20.2	78.2	1.6	20.5
Creosote and pitch ..	11.65	154	160	18.3	76.5	5.6	19.2
Fuel oil (C 86% ; H 12%) ..	14.0	184	194	14.1	74.5	11.4	15.9
Coal gas (560 B.Th.U.) ..	13.1	172	197	8.7	69.8	21.5	11.1
(35 cu. ft. coal gas/lb.)							

### CALCULATION OF THE COMPOSITION OF THE FLUE GASES FROM GASEOUS FUELS

The calculation of the composition of flue gases from a gas such as coal gas can be made on similar lines, but the fundamental combustion equations are more complicated and are utilised on a volume basis only. As an example, the combustion of methane is expressed by the equation :—



The interpretation of this equation in terms of waste gas analysis, as will be evident from previous explanations, is that when 1 cubic foot of methane is burned with air, it requires 2 cubic feet of oxygen (and consequently 9.56 cubic feet of air) and it forms as waste gas, 1 cubic foot of carbon dioxide, 2 cubic feet of water vapour and 7.56 cubic feet of nitrogen, all assumed to be measured at the same temperature.

Particulars of the combustion of some industrial gases are contained in Table 16.

TABLE 16

	Coal gas	Producer gas		Blast furnace gas
C.V.—B.Th.U./ cu. ft. gross ..	475	163	132	92
Cu. ft. air required for 1 cu. ft. gas (theoretical) .. ..	4.06	1.28	1.00	0.69
Products of combustion from 1 cu. ft. of gas :	cu. ft.	cu. ft.	cu. ft.	cu. ft.
CO <sub>2</sub> .. .. .	0.50	0.36	0.34	0.38
H <sub>2</sub> O .. .. .	0.97	0.18	0.12	0.02
N <sub>2</sub> .. .. .	3.27	1.53	1.34	1.14
Total .. .. .	4.74	2.07	1.80	1.54
Theoretical CO <sub>2</sub> content as analysed, i.e. on dry gas, per cent. .. .. .	13.3	19.2	20.5	24.9

## METHOD OF CALCULATION

Taking as an example the analysis of horizontal retort gas from Chapter II (Table 8), the combustion reactions and their resultant products can be summarised as in Table 17, noting that each cubic foot of oxygen required is accompanied by 3.78 cubic feet of nitrogen.

TABLE 17

Constituent	Combustion reaction	Cu. ft. per cu. ft. of constituent			
		O <sub>2</sub> req.	N <sub>2</sub> (from air)	CO <sub>2</sub> formed	H <sub>2</sub> O formed
Oxygen—O <sub>2</sub> .. ..	Assists in burning the combustible	—	—	—	—
Carbon dioxide—CO <sub>2</sub>	None	—	—	—	—
Hydrocarbons—C <sub>m</sub> H <sub>n</sub> , taken as C <sub>3</sub> H <sub>8</sub>	2C <sub>3</sub> H <sub>8</sub> + 9O <sub>2</sub> = 6CO <sub>2</sub> + 8H <sub>2</sub> O	4.5	17.01	3.0	3.0
Carbon monoxide—CO	2CO + O <sub>2</sub> = 2CO <sub>2</sub>	0.5	1.89	1.0	—
Hydrogen—H <sub>2</sub> ..	2H <sub>2</sub> + O <sub>2</sub> = 2H <sub>2</sub> O	0.5	1.89	—	1.0
Methane—CH <sub>4</sub> ..	CH <sub>4</sub> + 2O <sub>2</sub> = CO <sub>2</sub> + 2H <sub>2</sub> O	2.0	7.56	1.0	2.0
Nitrogen—N <sub>2</sub> .. ..	None	—	—	—	—

The calculation in cu. ft./100 cu. ft. of gas then proceeds as follows :—

Constituent	Per cent. by vol.	O <sub>2</sub> required for combus- tion cu. ft.	Products cu. ft.		
			CO <sub>2</sub>	H <sub>2</sub> O	N <sub>2</sub>
O <sub>2</sub> .. ..	0.4	—0.4	—	—	— 1.5
CO <sub>2</sub> .. ..	2.0	—	2.0	—	—
C <sub>3</sub> H <sub>8</sub> taken as					
C <sub>3</sub> H <sub>8</sub> .. ..	3.6	16.2	10.8	10.8	61.3
CO .. ..	8.0	4.0	8.0	—	15.1
H <sub>2</sub> .. ..	52.0	26.0	—	52.0	98.3
CH <sub>4</sub> .. ..	30.0	60.0	30.0	60.0	228.8
N <sub>2</sub> .. ..	4.0	—	—	—	4.0
Volume per 100 cu. ft. of town gas .. .. .. cu. ft.		105.8	50.8	122.8	404.0

The oxygen is provided from air ; thus to provide 105.8 volumes of oxygen

$$105.8 \times \frac{100}{20.9} = 506.2 \text{ volumes of air are required.}$$

In other words, the air required for the combustion of 100 volumes of gas is 506.2 volumes.

Thus 1 cubic foot of gas requires theoretically for combustion 5.062 cubic feet of air.

The products from 100 cubic feet of gas and 506.2 cubic feet of air will be :—

							Volume calc. at 60° F. and 30" bar., the water assumed to be uncondensed. cu. ft.	Per cent. by volume.
CO <sub>2</sub> .. ..	..	..	..	..	..	..	50.8	8.8
H <sub>2</sub> O (steam) .. ..	..	..	..	..	..	..	122.8	21.3
N <sub>2</sub> .. ..	..	..	..	..	..	..	404.0	69.9
Total .. ..	..	..	..	..	..	..	577.6	100.0

The volume of products formed when 1 cubic foot of this gas is completely burnt with 5.062 cubic feet of air is therefore 5.776 cubic feet at 60° F. and 30 inches bar.

If the gases are cooled below the dew point water vapour will condense and the volume will be less. The products for purposes of analysis are therefore generally expressed on the dry basis without the water vapour.

The volume of products from 100 cubic feet of gas is therefore on the dry basis :—

CO <sub>2</sub> .. ..	50.8 cubic feet
N <sub>2</sub> .. ..	404.0
Total	454.8

The volume of products of combustion (dry) from 1 cubic foot of this sample of town gas with 5.062 cubic feet of air is therefore 4.548 cubic feet.

The percentage of CO<sub>2</sub> as analysed from this gas is given by :—

$$50.8 \times 100 \quad 11.2 \text{ per cent.}$$

## CHEMICAL PRINCIPLES OF COMBUSTION AND GASIFICATION

The calculation can be extended to include excess air in the same way as for coal in the example previously given.

### THE CHEMISTRY OF GASIFICATION

A fuel bed may be shallow or deep. If shallow, the carbon tends to burn wholly to carbon dioxide. The carbon dioxide first formed, or water vapour present in the incoming air, may be brought into contact with red-hot carbon, in which event certain further reactions occur, to promote which is the function of gasification. A deep fuel bed naturally offers the optimum conditions for these further reactions, though they may occur to a lesser extent in the shallow fuel bed of the boiler furnace. A deep fuel bed of the order of 3 feet or more is used for gasification for the manufacture of producer gas.

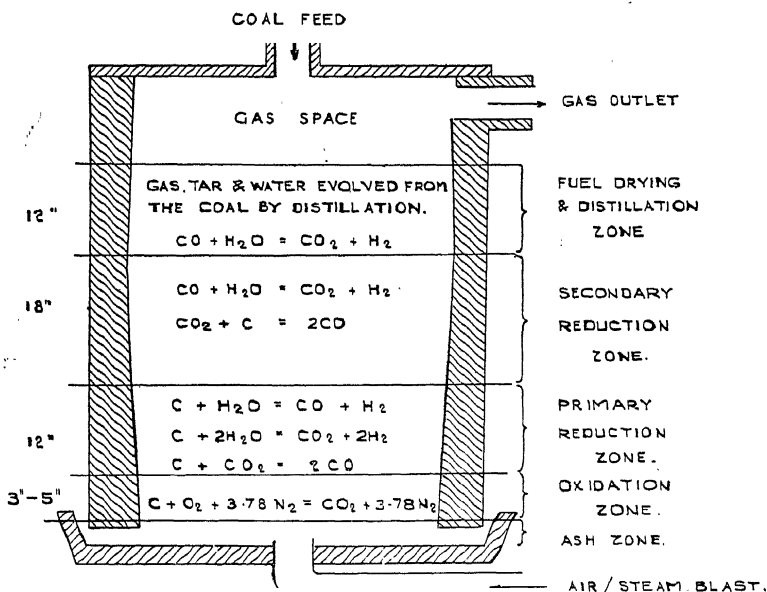


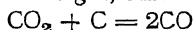
FIG. 13. Chemical reactions in a deep fuel bed.

It is convenient to consider these reactions as confined to several zones (Fig. 13) though they overlap into the zones above and below. The gases passing into the base of the fuel bed may comprise air and steam and when reacting in the fuel bed give rise to chemical and thermal effects of great importance. The significance of the thermal reactions here referred to will be explained in Chapter V.

The ash zone at the base of the fuel bed serves to protect the grate from the intense heat and to distribute the air and steam over the bed.

Above this is the oxidation zone at which the reaction  $\text{C} + \text{O}_2 = \text{CO}_2$  takes place, free oxygen disappearing about 4 or 5 inches above the top of the layer of ash. This reaction generates heat (see Chapter V) and provides practically the whole of the heat available in the fuel bed that is required for the subsequent gasification reactions.

The  $\text{CO}_2$ , accompanied by nitrogen and steam, travels upwards into the reducing zone. Here a reaction between  $\text{CO}_2$  and carbon takes place resulting in the production of the combustible gas, carbon monoxide:—





This reaction absorbs heat. The extent to which it occurs depends on the temperature and on the time available. Equilibrium values which are only attained by sustained contact in the absence of other disturbing factors, are given in Table 18.

TABLE 18

Temperature		Composition of gas in contact with carbon at equilibrium	
° C.	° F.	CO <sub>2</sub> —per cent.	CO—per cent.
600	1,112	68.5	31.5
700	1,292	36.6	63.4
800	1,472	13.7	86.3
1,000	1,832	0.6	99.4
1,200	2,192	0.06	99.94

To set up equilibrium conditions may require a very long time, depending on the velocity of the chemical reaction. Since the velocity of chemical reactions is, in general terms, approximately doubled by a rise of 10° C. (18° F.), at lower temperatures nothing like the equilibrium would be attained with industrial times of contact; since the initial gas is CO<sub>2</sub> the mixture at 600°–1,000° C. would be richer in CO<sub>2</sub> than those given in Table 18. The velocity of the reaction is considerably greater at the higher temperatures, and the effect of temperature on the reaction velocity is probably more important than its effect on the equilibrium.

At higher temperatures of the order of 1,200° C., the reaction might go substantially to completion, but owing to the influence of other factors this can but rarely be observed. Since any CO<sub>2</sub> present in producer gas represents waste of fuel—the carbon being completely burnt to CO<sub>2</sub> in the gas producer instead of in the furnace where the producer gas is burnt—the temperature in the gasification zones should clearly be high.

Steam also reacts with carbon in the reduction zone either :—



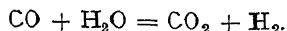
or



Reaction (1) occurs almost exclusively at and above 1,000° C. and reaction (2) at 600° C. Between these temperatures both reactions occur simultaneously, the extent of reaction (1) increasing as the temperature rises, until it predominates. To avoid the production of CO<sub>2</sub> in the producer, the temperature in the reduction zone should clearly be high for gasification. Both of these reactions absorb heat.

The importance of a high temperature in the reduction zone is thus clearly shown. The necessary heat is provided wholly by the combustion reaction in the oxidation zone, and this heat has (a) to heat up the fuel in the reduction zones to the necessary temperature, and (b) to compensate for the absorption of heat caused by the reactions in the reduction zone.

Higher still in the fuel bed, the reduction reaction  $\text{CO}_2 + \text{C} = 2\text{CO}$  and the reactions between carbon and steam still continue slowly and are now accompanied by a reaction between CO and still undecomposed steam, known as the water gas equilibrium reaction :—



This reaction persists until the temperature drops to a dull red heat and is responsible for the production of some additional CO<sub>2</sub> in the producer gas.

All the reactions are equilibrium reactions, and under suitable conditions may proceed in the reverse direction to those here indicated. At any tem-

perature there is an equilibrium composition of the substances taking part in the reaction which is the more closely approached as time and temperatures are adequate; an example was given when dealing with the reaction  $C +$

At the top of the fuel bed, the fresh fuel (if coal is used) is distilled, permanent gases, water and tar being added to the gases leaving the fuel bed. Any water contained in the incoming fuel is evaporated at this stage.

These reactions are summarised in Fig. 13; their thermal effects are summarised in Fig. 17.

The quantity of steam used has an important influence on the reactions. This will be discussed in detail in the chapter on gas producers (Chapter XVII). It is obviously important that as much of the steam as possible shall be converted into gas, and in its conversion into gas shall produce as little  $CO_2$  and as much  $CO$  and  $H_2$  as possible. The effect of temperature on the reaction between steam and carbon under practical conditions is well illustrated by experimental figures given in Table 19.

TABLE 19. THE ACTION OF STEAM ON CARBON (HARRIES)

Temperature		Steam decomposed Per cent.	Analysis of gases—Per cent.		
° C.	° F.		$CO_2$	$CO$	$H_2$
674	1,245	8.8	29.8	4.9	65.2
758	1,396	25.3	27.0	7.8	65.2
838	1,540	41.0	22.9	15.1	61.9
954	1,750	70.2	6.8	39.3	53.5
1,010	1,850	94.0	1.5	49.7	48.8
1,125	2,057	99.4	0.6	48.5	50.9

The net results of the various reactions here described can be followed experimentally. The course of the changes which take place has been determined by examination of gas samples withdrawn from closely adjacent sampling points in the fuel bed of a coke-fired producer, Table 20 (adapted from results quoted by Haslam and Russell). Table 30 (Chapter VI) is also of interest in this connection, though referring to the action of air only, steam not being admitted.

TABLE 20. COMPOSITION OF GASES IN FUEL BED OF A GAS PRODUCER

Zone	Height of sampling point above grate in.	Composition of gases—Per cent. by vol.				
		$O_2$	$CO_2$	$CO$	$H_2$	$H_2O$
Ash .. ..	5	18.4	—	—	—	13.4
Oxidation .. ..	7.5	nil	17.6	2.8	—	13.1
Primary reduction	10	—	11.5	12.2	—	12.2
" "	15	—	7.2	20.4	4.0	8.1
" "	20	—	5.9	23.3	7.0	4.7
Secondary reduction	30	—	4.6	25.6	8.2	3.5
Top of fuel bed ..	40	—	4.7	27.0	8.5	2.7

The final results of the gasification process are given in Chapter II, Table 9, under producer gas. If, as in some older producers, air only is blown through the fuel bed, the resulting gas is similar to blast furnace gas, though containing less  $CO_2$  and proportionately more  $CO$ .

## CHAPTER V

## HEATING EFFECTS IN COMBUSTION AND GASIFICATION

The measurement of heat—Specific heat—Excess air for combustion and its effects—Heat losses in flue gases—Thermal effects in gasification—Ignition temperatures—Ignition limits—Flame speed—Flame temperatures.

## THE MEASUREMENT OF HEAT

**Q**UANTITY of heat is defined as, and measured by, the amount of heat that is required to raise a given weight of water through a given temperature range.

On the pound-Fahrenheit scale heat is measured in terms of British Thermal Units (B.Th.U.). One B.Th.U. is the quantity of heat required to raise 1 lb. of water through 1° F. Since the heat capacity of water varies slightly at different temperatures the more accurate definition of the heat unit is the amount of heat required to raise 1 lb. of water from 60° to 61° F. Another definition is the "mean B.Th.U." which is  $\frac{1}{180}$  of the sensible heat required to raise 1 lb. of water at 32° F. to 212° F. without vaporisation.

On the kilogram-centigrade scale 1 large calorie (known as 1 kg. cal.) is the amount of heat required to raise 1 kilogram of water from 15° C. to 16° C. This is often written "K.C.U."

The pound-centigrade unit sometimes used is the amount of heat required to raise 1 lb. of water 1° C. It is denoted "C.H.U."

The relationship between these units is as follows :—

$$\begin{aligned} 1 \text{ kg. cal.} &= 3.968 \text{ B.Th.U.} \\ 1 \text{ C.H.U.} &= 1.8 \text{ B.Th.U.} \\ 1 \text{ B.Th.U.} &= 0.252 \text{ kg. cal.} \\ &= 0.555 \text{ C.H.U.} \end{aligned}$$

Thus if the calorific value of a coal is given as 14,000 B.Th.U. per lb. this means that if the whole of the heat generated by the combustion of 1 lb. of the coal under consideration were usefully employed in heating water, it would serve to raise 14,000 lb. of water through 1° F., or alternatively (for example) 140 lb. of water through 100° F. Thus,

lb. of water heated  $\times$  °F. temperature rise = B.Th.U. given to the water.

Heat, work and energy are interchangeable, and it is useful to note here certain other conversion factors, all in terms of British Thermal Units.

Unit.	Equivalent B.Th.U
1 foot-pound .. .. .	.. 0.001285
778 foot-pounds .. .. .	.. 1.0
1 therm. . . . .	.. 100,000
1 kilowatt-hour (kWh) . . .	.. 3,413
1 horse-power hour (H.P.H.)	.. 2,546

## PHYSICAL CONSTANTS

Figures of physical constants required in fuel technology can be found in "Technical Data on Fuel," edited by H. M. Spiers and published by the British National Committee of the World Power Conference. Only such figures are published here as are required for an explanation of the subject.

## HEATING EFFECTS IN COMBUSTION AND GASIFICATION

### DATUM CONDITIONS

In making thermal calculations it is necessary to select a basis temperature from which to start. For steam tables (Chapter VII) this temperature is 32° F. (0° C.) on the ground that below this temperature water solidifies and 32° F. is the lowest temperature at which water exists in the liquid state.

For most combustion calculations 60° F. is used and this temperature will be used in this book as the basis for this purpose.

### SPECIFIC HEAT

It is found experimentally that the quantity of heat required to raise 1 lb. of solid, liquid and gaseous substances through a given temperature range is not the same for each substance. Each substance is said to have its own particular specific heat, and this is defined with reference to water as the standard substance. Thus the specific heat of a substance is the number of B.Th.U. required to raise 1 lb. of the substance through 1° F., or alternatively the number of kg. cal. required to raise 1 kilogram through 1° C. The actual figure will be the same whichever system is used. By the definition of a heat unit, the specific heat of water at 15° C. or 60° F. is taken as 1.

The specific heats of solids and liquids are always given on a weight basis. The specific heat of a gas may be given on a weight basis, but for the purpose of calculation it is usually more convenient to give it on a volume basis. The volume basis will be adopted in this book. The specific heat of a gas, therefore, is expressed as the quantity of heat required to raise 1 cubic foot of the gas through 1° F.; or on the metric system, the quantity of heat required to raise 1 cubic metre of the gas through 1° C.; the figures on the two systems are different. It is, of course, necessary to state the temperature and pressure of the gas when quoting a specific heat on the volume basis.

The value of the specific heat of a substance varies to some extent with the temperature. Among examples of specific heats may be cited those in Table 21, which are given solely to illustrate the wide differences existing in this respect between various substances.

All gases have two specific heats, dependent upon whether they are being heated while keeping the pressure constant or the volume constant. Heating at constant pressure is far commoner than heating with constant volume; specific heats at constant volume are not generally needed in boiler and furnace combustion practice and are not therefore discussed here.

TABLE 21

Mean specific heats on a weight basis:—

						Specific heat	Over the temperature range	° C.
Water	..	..	..	..	..	1.00	60-61	15.6-16.1
Aluminium	..	..	..	..	..	0.216	32-212	0-100
Copper	..	..	..	..	..	0.102	32-600	0-315
Iron	..	..	..	..	..	0.128	32-750	0-399
Firebrick	..	..	..	..	..	0.23	32-1,800	0-982
Coal	..	..	..	..	..	0.25	32-300	0-149
Coke	..	..	..	..	..	0.26	80-750	26.7-399
Steam (at constant pressure)	..	..	..	..	..	0.45-0.52	212-600	100-315
Air (constant pressure)	..	..	..	..	..	0.24	32-500	0-260
Flue gases of average composition, dry (constant pressure)	..	..	..	..	..	0.24	212-600	100-315

Specific heats (on a volume basis) at a temperature of 0° C. (32° F.) unless otherwise stated, at constant pressure :—

	B.Th.U./cu. ft./° F. for gas measured at 60° F. and 30"	Kg. cal./cu. m./° C. for gas measured at 0° C. and 760 mm.
Air (dry) .. .. .	·0183	0·310
Carbon monoxide and nitrogen (dry) ..	·0184	0·311
Hydrogen (dry) .. .. .	·0181	0·306
Carbon dioxide (dry) .. .. .	·0226	0·384
Water vapour .. .. .	·0210	0·356
Mixed products of combustion (typical figures):— at 0° C.	·0191	0·324
Do. do. do. 0–300° C.	·0194	0·329
Do. do. do. 0–1,000° C.	·0207	0·352
Do. do. do. 0–1,500° C.	·0214	0·363

To illustrate the use of specific heat by examples, suppose it is required to calculate the heat required to heat 1 cwt. of firebrick from 60° F. to 1,780° F.

$$\begin{aligned}\text{Heat required} &= 112 \times (1,780 - 60) \times 0\cdot23 \\ &= \text{lb.} \times \text{temp. rise} \times \text{sp. ht.} \\ &= 44,310 \text{ B.Th.U.}\end{aligned}$$

Similarly, if it be required to calculate how much heat would be obtained if 750 cubic feet (measured at 60° F. and 30 inches) of products of combustion were cooled from 300° F. to 220° F., the figures would be :—

$$\begin{aligned}750 \times (300 - 220) \times 0\cdot0192 \\ \text{cubic feet} \times \text{temp. fall} \times \text{sp. ht.} \\ = 1,152 \text{ B.Th.U.}\end{aligned}$$

Thus, whilst for water,

$$\text{lb. weight} \times \text{temp. rise in } ^\circ\text{F.} = \text{B.Th.U.},$$

for all other materials,

$$\text{lb. weight} \times \text{temp. rise in } ^\circ\text{F.} \times \text{sp. ht. by weight} = \text{B.Th.U.}$$

and for gases, if using volumes,

$$\text{volume in cubic feet} \times \text{temp. rise in } ^\circ\text{F.} \times \text{sp. ht. by volume} = \text{B.Th.U.}$$

In accurate work it is necessary to take account of the variation of the specific heat with temperature and to use the mean specific heats between the temperatures involved. An example of this is given above when dealing with the specific heat of mixed products of combustion. Thus it may be required to calculate the heat above 60° F. remaining in 200 cubic feet of flue gases measured at 60° F. and 30 inches (a) at 575° F. (301° C.) and (b) 1,835° F. (1,002° C.). The volume of flue gas obtained from the combustion of 1 lb. of coal under good conditions in a boiler plant is approximately 200 cubic feet measured at 60° F.

(a) Gases at 575° F.

$$\begin{aligned}\text{Heat content} &= 200 \times (575 - 60) \times 0\cdot0194 \\ &\quad \text{cubic feet} \times \text{temp. rise} \times \text{sp. ht.} \\ &= 1,998 \text{ B.Th.U.}\end{aligned}$$

(b) Gases at 1,835° F.

$$\begin{aligned}\text{Heat content} &= 200 \times (1,835 - 60) \times 0\cdot0207 \\ &\quad \text{cubic feet} \times \text{temp. rise} \times \text{sp. ht.} \\ &= 7,348 \text{ B.Th.U.}\end{aligned}$$

## EXCESS AIR REQUIRED FOR COMBUSTION

When dealing with the chemistry of combustion, all calculations were made on the assumption that the fuels were burnt completely with the theoretical amount of air. If more than this amount of air is used, the foregoing calculations can be readily adjusted by adding the additional oxygen (20.9 per cent. of the extra volume of air) and nitrogen (79.1 per cent.) to the gases derived from combustion.

In practice complete combustion cannot be secured unless more than the theoretical amount of air is used. When burning solid fuel this is partly because of the difficulty of mixing the air and combustible volatiles above the fuel bed. It is also a consequence of the necessity of completing the combustion within the combustion space. Physical chemistry has shown that the speed of a chemical reaction is proportional to the concentration of the reactants. As the concentration of oxygen and combustible in the combustion space decreases from front to back of the combustion zone, it is necessary to add an excess of oxygen to keep the reaction rate high enough to complete the reaction before the gases leave the furnace.

The velocity of combustion depends upon the rate at which oxygen can reach the fuel. Scrubbing action due to the velocity of the gases promotes the interchange of  $O_2$  and  $CO_2$  with solid fuel surfaces and accelerates the combustion; this is indicated by the rapidity with which coal burns at a thin spot or over the end of a tuyere where the air velocity is great.

## HEAT LOSSES THROUGH EXCESS AIR IN FLUE GASES

It is important to recognise the effect of excess air in combustion. When gases are hot they possess what is known as "sensible heat," the amount of which (as has just been shown) is given in B.Th.U. by the product:—

Vol. of gases in cubic feet measured at 60° F.  $\times$  specific heat in B.Th.U. per cubic foot  $\times$  °F. rise in temperature.

The quantity of heat they contain is thus dependent on their specific heat and on their temperature, and, of course, it is directly proportional to their volume measured at standard temperature and pressure. When excess air is used the specific heat of the flue gases is not appreciably affected. The primary effect is to increase the volume of gases and thus directly to increase the loss of heat (see Table 22).

In the measurement of flue gas temperature, sometimes serious errors can be caused if precautions are not taken to shield the thermometer or thermocouple from the radiation effect of the surrounding flue walls or of cooler materials around it, e.g. economiser tubes. This effect is discussed in Chapter XXX.

## LATENT HEAT IN FLUE GASES

In addition to their sensible heat, flue gases contain also latent heat in the latent heat of condensation of the steam resulting from combustion or evaporated from the fuel. The subject of latent heat is dealt with in Chapter VII. It is there shown that the latent heat of steam depends on the pressure at which it is generated or condensed. For condensation from flue gases, it may be taken as 1,055 B.Th.U. per lb. or 49.35 B.Th.U. per cubic foot of water vapour. This latent heat, which must be added to the sensible heat, is thus given by

$$(a) \text{ Wt. of steam (in lb.) } \times 1,055 \text{ B.Th.U.}$$

or

$$(b) \text{ Vol. of steam (in cubic feet) } \times 49.35 \text{ B.Th.U.}$$

All gas and water vapour volumes are taken for the purpose of these calculations as being at 60° F. and 30 inches barometric pressure.

### EFFECT OF EXCESS AIR ON CO<sub>2</sub> CONTENT OF FLUE GASES

The dilution with air proportionately decreases the CO<sub>2</sub> content of the waste gases, because the same volume of CO<sub>2</sub> is contained in a greater volume of other gases. Conversely, it is clearly possible from a knowledge of the CO<sub>2</sub> content and the temperature of the gases to deduce the quantity of heat carried away by hot products of combustion.

### TOTAL HEAT LOSSES IN FLUE GAS

The heat losses in flue gas are thus seen to be made up of several quantities, some of which are controllable and others outside our control.

- (a) Sensible heat of gases derived from combustion.
- (b) Sensible heat of excess air.
- (c) Latent heat of steam arising from moisture in fuel.
- (d) Latent heat of steam arising from combustion of the hydrogen in the fuel.
- (e) Sensible heat of steam arising from moisture in fuel.
- (f) Sensible heat of steam from hydrogen in fuel.

To the extent that the outlet temperature of the gases from the plant can be controlled through more efficient operation or through the installation of more efficient apparatus (e.g. economisers in a boiler plant, or regenerators or waste heat boilers in a furnace) all of the foregoing sources of loss are partly controllable, except (c) and (d). (c) is controllable if the moisture content of the fuel can be reduced. No control can be exercised over (d). The maximum of control can be exercised over (b).

To indicate the magnitude of these losses, the same coal will be considered that was the subject of the calculation of flue gas analysis in Chapter IV. This coal contained 9 per cent. of moisture and 5 per cent. of hydrogen. Neglecting SO<sub>2</sub>, the composition of the flue gases with 50 per cent. excess air, and taking all volumes at 60° F., was :—

	Cu. ft./lb. coal
CO <sub>2</sub> .. ..	23·3
N <sub>2</sub> .. ..	104·1
Excess air .. ..	65·7
Steam from moisture in coal	
Steam from hydrogen in coal	assumed uncondensed.
	204·4

If the gases are leaving the system at 560° F., the following calculations may be made to indicate the relative effects of the losses (a) to (f).

- (a) Volume of gases (CO<sub>2</sub> + N<sub>2</sub>) .. .. 127·4 cubic feet  
 Sp. ht. over the range 560°–60° F. .. .. 0·0195  
 Heat in gases =  $127·4 \times 0·0195 \times (560-60)$   
                   = 1,242 B.Th.U./lb. coal.
- (b) Volume of excess air .. .. 65·7 cubic feet  
 Sp. ht. over the range 560°–60° F. .. .. 0·0196  
 Heat in excess air =  $65·7 \times 0·0196 \times (560-60)$   
                   = 644 B.Th.U./lb. coal
- (c) Volume of steam from moisture in fuel (calculated to 60° F. and 30 in. bar.) .. .. 1·9 cubic feet  
 Latent heat per cubic foot .. .. 49·35 B.Th.U.  
 Total latent heat =  $1·9 \times 49·35$   
                   = 93·8 B.Th.U./lb. coal

- (d) Volume of steam from combustion of hydrogen  
in fuel (calc. to 60° F. and 30 in. bar.) .. cubic feet  
Total latent heat =  $9.4 \times 49.35$   
= 463 B.Th.U./lb. coal
- (e) and (f) Total heat of superheated steam at  
560° F. at the pressure at which it exists in  
the flue gas, i.e.  $(14.7 \times 11.3/205.4 =)$   
0.8 lb. per sq. in. absolute (see Chapter VII),  
taken from steam tables .. .. . 1,317 B.Th.U. per lb.  
Since steam tables are based on 32° F. and the  
flue gas calculations are based on 60° F.,  
from this must be deducted the heat required  
for heating the condensed water from 32° to  
60° F. .. .. . 28 B.Th.U. per lb.

Total heat of water vapour above 60° F. 1,289 do.  
To obtain the sensible heat (or superheat) in  
the steam, the latent heat must be deducted  
also ; this is .. 1,055 do.

The sensible heat per lb. of steam is thus .. 234 B.Th.U.  
Since (Table 14) water vapour at 60° F. weighs  
0.0476 lb./cu. ft., the sensible heat per cubic  
foot is  $234 \times 0.0476 =$  .. .. 11.1 B.Th.U.  
Thus, for (e) sensible heat of steam =  $1.9 \times 11.1 =$  21.1 B.Th.U./lb. coal.

for (f) sensible heat of steam =  $9.4 \times 11.1 =$  104.3 B.Th.U./lb. coal.

These figures may be summarised thus, the coal being taken to have a calorific value of 14,000 B.Th.U./lb. :—

	B.Th.U./lb. of coal burnt	Per cent. of heat in coal
(a) Sensible heat of gases derived from combustion ..	1,242	8.87
(b) Sensible heat of excess air .. .. .	644	4.60
(c) Latent heat of steam from moisture in fuel ..	93.8	0.67
(d) Latent heat of steam from hydrogen in fuel ..	463	3.30
(e) Sensible heat of steam from moisture in fuel ..	21.1	0.15
(f) Sensible heat of steam from hydrogen in fuel ..	104.3	0.75
	2,568.2	18.34

Since (d) cannot be controlled, this calculation indicates that (a) and (b) are of much greater importance in promoting fuel economy than any other factors. Hence the importance of

- (1) Operating with the lowest amount of excess air that will avoid still greater losses due to unburnt combustible gases.
- (2) Operating with the lowest practicable outlet temperature from the system.

The result of a calculation of this character is shown in Fig. 14 for a bituminous coal and in Fig. 15 for coal gas.



# THE EFFICIENT USE OF FUEL

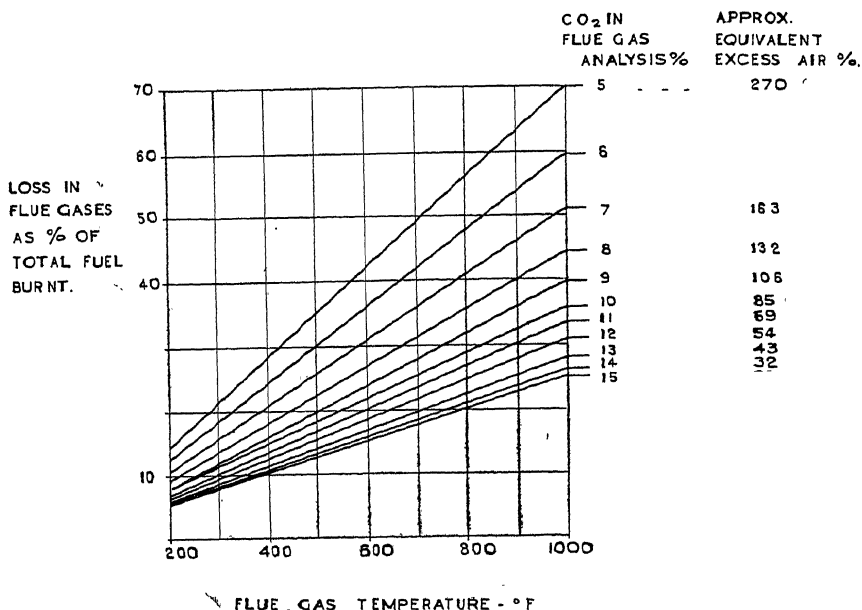


FIG. 14. Effect of temperature and excess air upon the loss of heat in flue gases from coal.  
(Reproduced from "Fuel Saving in the Iron and Steel Industry," by permission of the Iron and Steel Industrial Research Council.)

Table 22 further illustrates this point and gives the percentage of the heat of the incoming coal that is contained in the flue gases under various conditions.

The figures in Table 22 also show that the heat losses increase rapidly as the percentage of CO<sub>2</sub> falls and that 1 per cent. of CO<sub>2</sub> in the waste gases means more in terms of heat saved or lost for low CO<sub>2</sub> content than for high CO<sub>2</sub> content.

TABLE 22. HEAT LOSSES IN FLUE GASES FROM BITUMINOUS COAL

Per cent. CO <sub>2</sub> in flue gas	Per cent. excess air used	Heat losses at flue gas temperatures of		
		120° C. (248° F.) per cent.	170° C. (338° F.) per cent.	200° C. (392° F.) per cent.
15	25	8.9	11.2	12.6
14	33	9.2	11.6	13.1
13	43	9.5	12.1	13.7
12	55	9.9	12.7	14.4
11	68	10.4	13.4	15.3
10	88	11.0	14.3	16.3
9	103	11.7	15.3	17.5
8	128	12.5	16.6	19.0
7	160	13.7	18.3	21.0
6	206	15.2	20.5	23.6

The first two columns of this table indicate how the proportion of excess air used can be broadly ascertained for a bituminous coal from the flue gas analysis. These figures would not apply to other fuels.

## CALCULATION OF FLUE GAS LOSSES FROM SOLID FUELS

In practice it is usually found that the general characteristics of the coal used can be ascertained by simple methods of analysis, but the elementary composition (per cent. of C, H, O, etc., by weight) is not known. The  $\text{CO}_2$  content of the flue gases and their temperature can be determined on the plant. It is then required to calculate the flue gas loss.

An indication sufficient for practical purposes can be obtained from Fig. 14. For those who wish to follow the calculation in greater detail, the following method is used. The coal in use is assumed to be known to be similar to that containing 30 per cent. of volatile matter in Table 15. It is found by tests that the flue gases contain 10 per cent.  $\text{CO}_2$  and are at a temperature of  $500^\circ\text{F}$ . at the plant outlet. How much of the heat in the coal is lost?

From Table 15 :—

Theoretical air required per lb. coal .. .. .	140 cubic feet at $60^\circ\text{F}$ . and 30 inches pressure
Theoretical volume of wet waste gases per lb. coal ..	147 cubic feet at $60^\circ\text{F}$ . and 30 inches pressure
$\text{CO}_2$ content of wet waste gas ..	17.3 per cent...25.4 cubic feet per lb. coal
„ dry ..	18.6 per cent.
$\text{H}_2\text{O}$ content of wet waste gas ..	6.9 per cent...10.1 cubic feet per lb. coal

Volume of  $\text{CO}_2$  per lb. of coal .. .. . 25.4 cubic feet

Volume of total dry gas per lb. coal when  $\text{CO}_2$  content of dry gas is 10 per cent. .. . 254.0 „ „

Volume of water vapour .. .. . 10.1 „ „

Volume of total wet gas .. .. . 264.1

Volume of gases of combustion .. .. . 147.0

Volume of excess air .. .. . 117.1

Percentage of excess air .. .. .  $\frac{117.1}{140} \times 100 = 83.7$  per cent.

Mean specific heat of total flue gases between the temperatures  $60^\circ\text{F}$ . and  $500^\circ\text{F}$ . .. 0.0197 B.Th.U. per cubic foot.

Sensible heat content of gases per lb. coal =  $264.1 \times 0.0197 \times (500-60)$   
= 2,287 B.Th.U.

Latent heat content of gases per lb. coal =  $10.1 \times 49.35$   
= 498 B.Th.U.

Total heat content =  $2,287 + 498$   
= 2,785 B.Th.U.

If the calorific value of this coal is 14,400 B.Th.U. per lb. the loss of heat in the flue gases is

$$\frac{2,785}{14,400} \times 100 = 19.35 \text{ per cent.}$$

## CALCULATION OF FLUE GAS LOSSES FROM TOWN GAS

As an example of the method of dealing with gaseous combustion the flue

gas losses from town gas will be next discussed. The gas selected for calculation in Chapter IV yielded :—

	Per 100 cubic feet of gas	Per cubic foot of gas
CO <sub>2</sub>	50.8	0.508
H <sub>2</sub> O	122.8	1.228
N <sub>2</sub> ..	404.0	4.040
Total	577.6	5.776

(a) *Amount of Excess Air.* If more air than required is passing into a combustion chamber the process may be represented by the equation

Gas + air for combustion + excess air = Products of combustion + excess air

the amounts of each constituent being expressed by weight. The flue gas analysis is made on gas from which the water vapour has been condensed and thus, taking CO<sub>2</sub> as an example,

$$\text{CO}_2 \% = \frac{\text{cu. ft. CO}_2 \text{ produced per cu. ft. of gas} \times 100}{\left\{ \frac{\text{cu. ft. CO}_2 + \text{N}_2 \text{ produced}}{\text{per cu. ft. of gas}} \right\} + \left\{ \frac{\text{cu. ft. of excess air}}{\text{per cu. ft. of gas}} \right\}}$$

The excess air contains 20.9 per cent. of oxygen, so that for the example taken, if X be the cubic feet of excess air used per cubic foot of gas, an analysis of the flue gases will give

$$\begin{aligned} \text{CO}_2 \% &= \frac{0.508 \times 100}{0.508 + 4.04 + X} \\ \text{O}_2 \% &= \frac{0.209X \times 100}{0.508 + 4.04 + X} \end{aligned} \quad (2)$$

Both CO<sub>2</sub> per cent. and O<sub>2</sub> per cent. can be obtained from the Orsat analysis so that excess air can be expressed as :—

$$\text{From (1) } \text{CO}_2 (0.508 + 4.04 + X) = 0.508 \times 100$$

$$X = \frac{50.8}{\text{CO}_2} = 4.548$$

$$\text{From (2) } (4.548 + X) \text{ O}_2$$

$$4.548 \text{ O}_2$$

$$20.9X$$

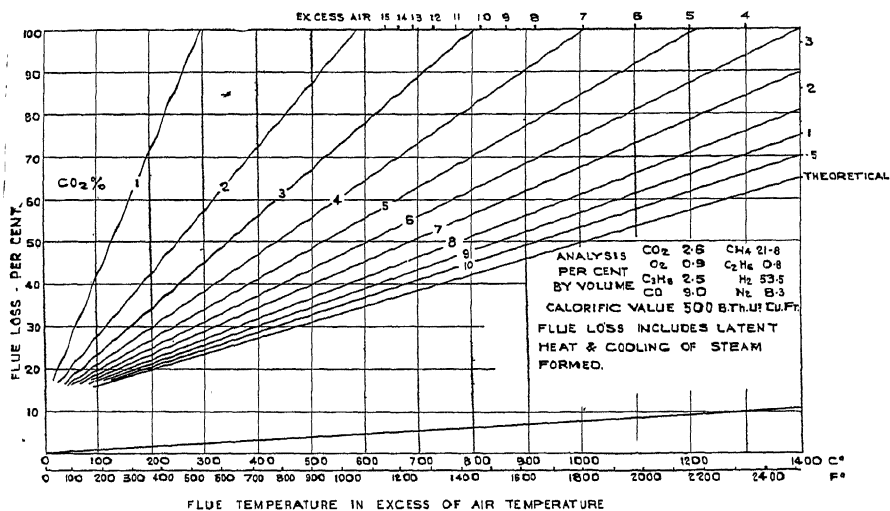
$$(20.9 - \text{O}_2)X$$

$$X = \frac{4.548 \text{ O}_2}{20.9 - \text{O}_2}$$

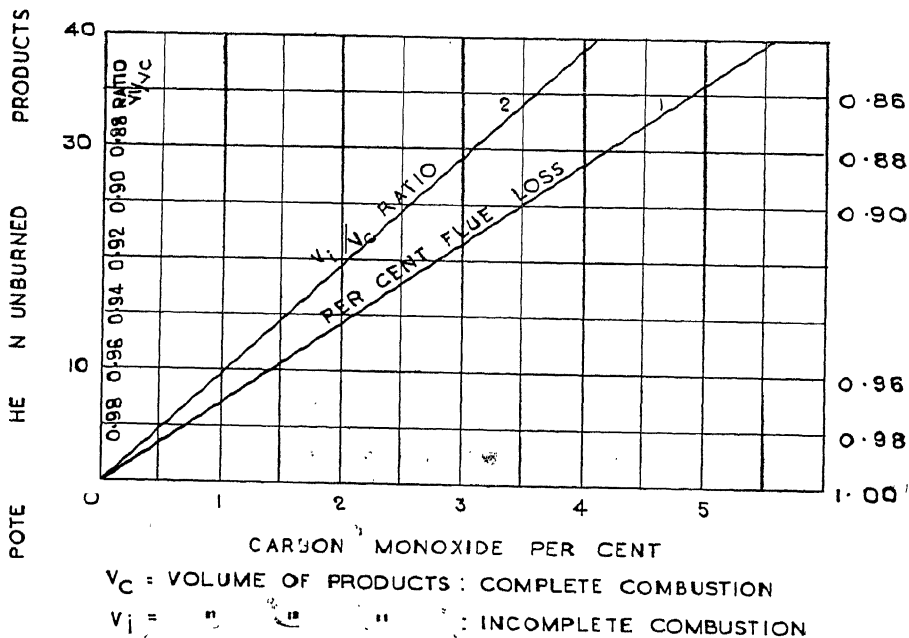
(b) *Incomplete Combustion.* If the air for combustion is insufficient, combustible gases are present in the flue. With small deficiency of air CO and H<sub>2</sub> are present, but as the air is further decreased complex partially oxidised hydrocarbons are formed which have an unpleasant smell. Thus incomplete combustion cannot be detected by smell until quite serious waste is occurring, and 2 to 3 per cent. of CO may be present before a distinct smell is noticeable. Frequently, however, local incomplete combustion occurs owing to a badly designed (or defective) combustion chamber and the products may have an odour before general incomplete combustion occurs.

Analysis shows that in the initial stages of incomplete combustion the ratio of hydrogen to carbon monoxide is 2 : 1, so that for every 1 per cent. carbon monoxide in the flue there is a potential heat loss due to incomplete combustion alone of 7 per cent. The importance of controlling combustion by flue gas

## HEATING EFFECTS IN COMBUSTION AND GASIFICATION



15. Effect of temperature and excess air upon the loss of heat in flue gases from town gas.



16. Diagram for calculation of flue gas loss in the presence of unburned gases.

analysis is at once apparent. This applies equally, of course, to the combustion of solid fuels.

(c) *Flue Losses.* The calculation of the heat content of the flue gases requires a knowledge of the flue gas analysis, the flue gas temperature and also the heat capacities of the various flue gas constituents. Fig. 15 is a flue loss chart for a typical gas of 500 B.Th.U. per cubic foot, the numbers on the curves indicating the percentage  $\text{CO}_2$  in the flue gases.

In practice, if combustion is complete it is only necessary to analyse for  $\text{CO}_2$ . Precautions are necessary in taking the sample to ensure that it is truly representative and that the sampling tube is not in a dead gas pocket (cf. Figs. 63 (a) and (b), Chapter VIII), and also that sufficient time has elapsed since the previous adjustment for steady conditions to be established. These are discussed in Chapter XXXII.

(d) *Flue Losses under Conditions of Incomplete Combustion.* For strictly accurate work these must be calculated from the flue gas analysis and the flue gas temperature, but for routine work this is tedious, and the curves in Fig. 16 are given for use with the flue loss diagram in Fig. 15 to enable the loss to be read off with reasonable accuracy. The method is as follows:—

- (1) Having obtained, by analysis, the CO percentage in the flue products and the flue gas temperature, read off the flue loss at theoretical  $\text{CO}_2$  (Fig. 15).
- (2) Multiply this value by the  $V_i/V_c$  ratio for the CO found (Curve 2, Fig. 16).
- (3) Read the potential heat in the unburned flue products for the CO found (Curve 1, Fig. 16).
- (4) The total flue loss is given by the sum of the results from (2) and (3).

Example: A carburising furnace has a flue temperature of  $925^\circ\text{C}$ ., and a flue gas analysis of  $\text{CO}_2$  10.5,  $\text{O}_2$  nil, CO 1.5 per cent. The air temperature is  $60^\circ\text{F}$ . What is the flue loss?

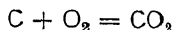
- (1) Flue loss (Fig. 15) at  $(925-15)^\circ\text{C}$ . and theoretical  $\text{CO}_2$  = 46 per cent.
- (2)  $V_i/V_c$  ratio (Fig. 16) at 1.5 per cent. CO = 0.94  
 $\text{flue loss} \times V_i/V_c = 46 \times 0.94 = 43.3$  per cent.
- (3) Potential heat in unburned flue products at 1.5 per cent. CO (Curve 1, Fig. 16) = 11 per cent.
- (4) Total flue loss =  $43.3 + 11$  = 54.3 per cent.

## THERMAL EFFECTS IN COMBUSTION

The thermal effect of the chemical reactions comprised in combustion is, of course, the fundamental basis of heat evolution. The simple facts that when a combustible burns it gives out heat, and that all combustibles do not evolve the same amount of heat per unit weight are accounted for by the thermochemistry of the reactions between carbon, hydrogen and oxygen.

The heat liberated by direct combustion between carbon and oxygen depends upon the form in which the carbon exists.

In industrial practice carbon may exist in three forms—graphite, amorphous carbon and in combination with other elements. What are considered to be the most reliable figures indicate that the calorific value (C.V.) of carbon differs in different industrial products as in Table 23. This refers to the reaction:—



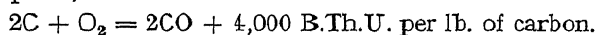
The thermal effect of other reactions of carbon must also depend in a minor degree upon the type of carbon that is reacting.

Under conditions of insufficient air supply the reaction will go only half-way

TABLE 23. C.V. OF CARBON AND HIGH-CARBON FUELS

<i>Carbon :</i>		B.Th.U./lb.
Carbon as graphite .. ..		14,130
Carbon in metallurgical coke		14,200
Carbon in horizontal retort coke		14,290
Carbon in vertical retort coke		14,380
Carbon in coal .. ..		14,450
<i>Fuels :</i>		
Ash-free, dry metallurgical coke		14,250
Ash-free, dry gas coke ..		14,400
Combustible matter in ashes ..		14,540 (B.S.I. figure)

and carbon monoxide is formed. The thermal value of the reaction, based on carbon as graphite, is :—



When the carbon monoxide thus formed is burnt to  $CO_2$  the net result is the same as if the carbon had been burnt directly to  $CO_2$ . The C.V. of carbon monoxide is an invariable figure for which the most reliable determinations are close to 10,150 B.Th.U. per lb. of carbon contained in the CO. The heat generated by the combustion of carbon to CO depends on the character of the carbon from which it is formed (Table 24).

TABLE 24. HEATS OF REACTION OF C, CO AND  $O_2$  IN B.Th.U. PER LB. CARBON

	Graphite	Metallurgical coke	Gas coke	Coal
C to CO .. ..	3,980	4,050	4,200	4,300
CO to $CO_2$ .. ..	10,150	10,150	10,150	10,150
C to $CO_2$ .. ..	14,130	14,200	14,350	14,450

Thus when carbon burns to CO there is a loss of 10,150 B.Th.U. per lb. of carbon, over 70 per cent. of the initial heat in the carbon, a fact which provides adequate reason for the need for obtaining complete combustion. In a gas producer, there is evolved some 4,000 B.Th.U. per lb. of carbon, and the remaining 10,150 B.Th.U. is generated when the CO is burnt as a constituent of the producer gas.

The combustion of hydrogen is complicated by the formation of water vapour. In furnace work, it is rare for the products of combustion to be cooled within the process to such a degree that the water vapour is condensed to liquid. When the water formed by combustion is not condensed within the system, as has been pointed out previously, heat is unavoidably lost amounting to 1,055 B.Th.U. per lb. of water not condensed (49.35 B.Th.U. per cubic foot of water vapour assumed measured at 60° F.).

Thus hydrogen and all fuels containing hydrogen have two calorific values. The higher or "gross" value assumes that all the water is condensed and cooled to atmospheric temperature. The lower or "net" value assumes that none of the water is condensed, and is obtained from the gross value by deducting 1,055 B.Th.U. per lb. of water condensed from the products of combustion by cooling to 60° F.

Combustion of hydrogen :—

$2\text{H}_2 + \text{O}_2 = 2\text{H}_2\text{O}$  (liquid) + 61,500 B.Th.U. per lb. of hydrogen  
and

$2\text{H}_2 + \text{O}_2 = 2\text{H}_2\text{O}$  (vapour) + 52,000 B.Th.U. per lb. of hydrogen.

Combustion of sulphur :—

$\text{S} + \text{O}_2 = \text{SO}_2$  + 4,000 B.Th.U. per lb. of sulphur.

When the combustion of a solid or liquid fuel containing oxygen is considered, since the oxygen present is assumed to be already combined with the hydrogen of the fuel an allowance of hydrogen equivalent to the oxygen present must be deducted from the effective weight of combustible hydrogen. This deduction is equal to one-eighth of the oxygen content, as has already been explained.

The calorific value of a coal may be calculated from its composition (see Table 1) by the following expression :—

$$\text{Gross C.V. (B.Th.U. per lb.)} = [\text{C per cent.} \times 14,450 + (\text{H per cent.} - \frac{\text{O per cent.}}{8}) \text{S per cent.} \times 4,000] / 100$$

For calculating the net C.V. of coal, the C.V. of 61,500 for available hydrogen must be replaced by 52,000.

This formula gives correct results for coal within about  $\pm 200$  B.Th.U. per lb., the discrepancy arising from the fact that formulæ of this character cannot take into account the heat of formation of the original material.

The calorific value of gases is determined and calculated by volume, and in British units is expressed in B.Th.U. per cubic foot measured at  $60^\circ \text{F. sat.}$

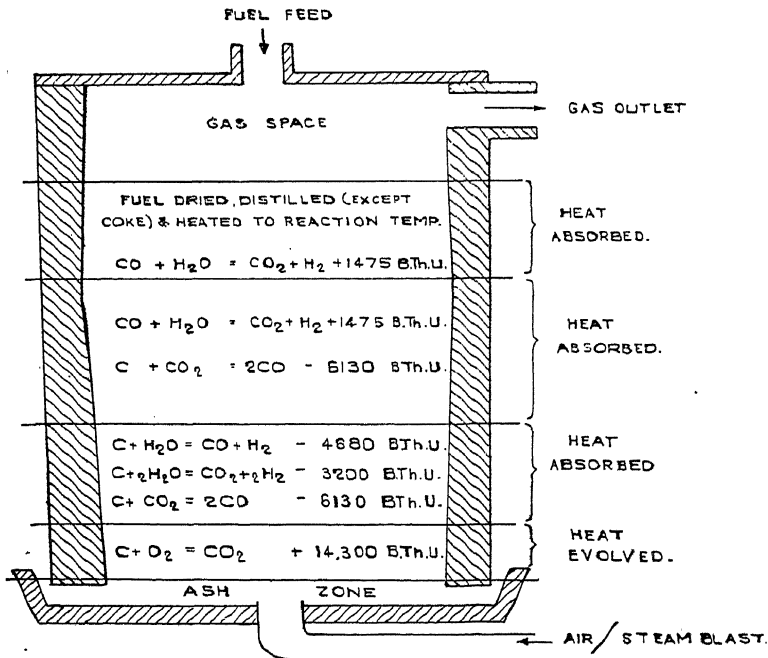


Fig. 17. Thermal effects in a deep fuel bed (gasification).

and 30 inches pressure. This subject was dealt with under "Coal Gas" in Chapter II, where a list of the C.V. of industrial gases under these conditions was given.

### THERMAL EFFECTS IN GASIFICATION

The chemical reactions which take place when a mixture of air and steam are blown into a deep fuel bed—and to a lesser extent in a shallow fuel bed—have been described in Chapter IV (Fig. 13). The importance of temperature in enabling the desired reactions to be effected with the necessary velocity was there emphasised. To obtain the required temperature in the reaction zones it is necessary to generate sufficient heat from the combustion of the fuel to compensate for all losses, including the heat absorbed in chemical reactions. A knowledge of the thermal effects in gasification is thus very necessary.

Fig. 17 indicates these thermal effects expressed as B.Th.U. per lb. of carbon taking part in the reaction. The thermal effect of some of these reactions depends on the condition of the carbon as has just been indicated. With one minor exception, all the reactions absorb heat except the direct combustion of the fuel at the base of the producer, as indicated by the "minus" sign.

### IGNITION TEMPERATURE

No substance will burn until it has been raised to a temperature at which its reaction with oxygen is sufficiently rapid. Visible ignition is preceded by a temperature range in which slow combustion is occurring. This was illustrated by the behaviour of coal during storage (Fig. 12, Chapter III).

The ignition temperatures in air of a number of fuels are given in Table 25. These temperatures are not absolute, but depend to a greater or lesser extent on the conditions.

TABLE 25. IGNITION TEMPERATURES IN AIR

	°C.	
Bituminous gas coal ..	370	700
Ordinary bituminous coal	400-425	750-800
Welsh steam coal ..	470	880
Anthracite .. ..	500	930
Coke (soft) .. ..	425-500	800-930
Coke (hard) .. ..	500-650	930-1,200
Hydrogen .. ..	580-590	1,080-1,095
Carbon monoxide ..	644-658	1,191-1,216
Methane .. ..	650-750	1,200-1,380
Ethane .. ..	520-630	970-1,170
Ethylene .. ..	542-547	1,008-1,016
Benzene .. ..	740	1,365
Toluene .. ..	810	1,490
Ethyl alcohol ..	558	1,036
Cylinder oil ..	417	783
Gas oil .. ..	336	638
Kerosene .. ..	295	562

### IGNITION LIMITS

If a trace of gas is added to air and the mixture is brought into contact with a flame or an electric spark the mixture will not ignite. If, for example, the mixture consisted of 3 per cent. of coal gas with air there will be no ignition. As the percentage of coal gas is increased ignition does not occur until a limiting value is reached of 5.3 per cent. This is known as the lower limit of inflam-



mability and the mixture will ignite. Increasing the proportion of gas renders the mixture more readily ignitable, but as the percentage of gas is still further increased the inflammability becomes less until at and over 31 per cent. of gas there is again no ignition. This is known as the higher limit of inflammability. The limits of some industrially important gases will be found in Table 26.

TABLE 26. LIMITS OF INFLAMMABILITY IN AIR

Substances	Lower limit per cent. by volume	Higher limit per cent. by volume
Coal gas .. .. .	5.3	31
Benzene .. .. .	1.4	7.5
Blast furnace gas .. .. .	35.0	74
Carbon monoxide .. .. .	12.5	74
Hydrogen .. .. .	4.0	75
Methane .. .. .	5.0	15

### FLAME SPEED AND ITS EFFECT ON INDUSTRIAL BURNERS

When an inflammable mixture of gas and air in a tube is ignited the resultant flame travels through the mixture, at first with a constant velocity and subsequently under suitable conditions with an accelerating velocity. This occurs particularly in the presence of turbulence. Conditions of deterioration may ultimately be reached in which the flame speed becomes accelerated to a very high velocity of the order of 3,300–6,600 feet per second.

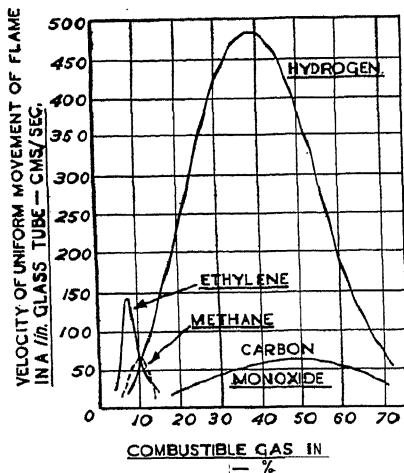


FIG. 18. Speed of flame propagation for various mixtures of air with combustible gases. Summary of Professor Wheeler's results.

(Reproduced by permission of the publishers (Walter King) from "Theory of Gas Heating," by Peter Lloyd.)

(ii) for the burning gases to be blown off the burner head when the forward velocity of the combustible mixture exceeds the speed of flame propagation. The high flame speed of hydrogen assists stability on the burner head.

### FLAME TEMPERATURE

The temperature which is attained by a gas flame (or the flame of the volatile matter from coal) depends upon several factors.

- (a) The calorific value of the gas or vapour.

The heat supplied to the flame is mainly obtained from the combustion of the fuel. Since the steam formed by combustion will leave the flame uncondensed the calorific value that is available is represented by the net C.V. and not the gross C.V.

- (b) The sensible heat of the air and gas.

If either the air or the gas is preheated the sensible heat thus given to the reacting substances will be added to the heat derived from combustion.

These two items together (minus item (e) below), make up the total heat input into the flame.

- (c) The quantity and specific heat of the products of combustion.

The heat derived from the combustion of the gas is used for heating the products of combustion up to the flame temperature. The mass of the products of combustion multiplied by their total heat capacity over the temperature range represents the heat thus utilised. Since the specific heat of gases varies with the temperature, precise knowledge of the specific heat at high temperatures is required for the determination of the "theoretical flame temperature" so defined.

Clearly the less the heat required by the products of combustion, the greater will be the flame temperature. Inerts in the gas or any gases added to the flame which do not take part in the combustion must reduce the flame temperature.

- (d) Excess air.

Any excess air used in the flame will add heat if the air is preheated but will on balance remove heat from the flame since the excess air must itself be raised to flame temperature.

- (e) Dissociation.

The combustion reactions  $C + O_2 = CO_2$  and  $2H_2 + O_2 = 2H_2O$  are reversible at high temperatures, which in simple language means that at very high temperatures these reactions do not go to completion. Consequently a proportion of the heat of combustion must be subtracted from item (a) in accordance with the extent to which the combustion is incomplete. The higher the temperature of the flame the more incomplete will be the combustion in the flame. This effect can be ignored generally in practical applications of combustion.

- (f) Radiation from the flame.

Theoretical flame temperatures are calculated on the assumption that no heat is radiated from the flame. This is obviously incorrect since the purpose of producing a flame is that it shall do work, i.e. that heat shall be radiated from it. For this reason flame temperatures in practice are always lower than the theoretical flame temperatures calculated according to the factors

(a) to (e)

Neglecting the loss of heat from radiation under (f) the theoretical flame temperature is thus given by the fraction :—

$$\frac{\text{Net C.V. of gas} + \text{sensible heat of gas and air} - \text{heat loss by dissociation}}{\text{Vol. of products} \times \text{sp. ht. per unit volume at constant pressure}}$$

Calculations of flame temperature made upon these principles, the accuracy of which has been confirmed experimentally, indicate that with the exception of acetylene which is somewhat above the range and with a slight exception for hydrogen and carbon monoxide all gases containing the normal quantity of inerts up to, say, 10 per cent. or so have virtually the same flame temperature. This is because any additional heat given to the flame by increased

calorific value is counter-balanced by a greater quantity of products of combustion.

Flame temperature has a special significance as governing the thermal efficiency of the evolution of heat from flames. This efficiency is measured by the ratio of temperature difference between the flame temperature ( $t_1$ ), and that of the issuing flue gases ( $t_2$ ), i.e.

Some figures are given in Table 27.

TABLE. 27. FLAME TEMPERATURE OF GASES AND VAPOURS

	Gross C.V. of gas	Per cent. inerts in gas	Flame temperature	
			° F.	° C.
Acetylene .. ..	1,500	nil	4,230	2,340
Ethylene .. ..	1,560	nil	3,700	2,040
Natural gas .. ..	1,220	1	3,645	2,030
" " " " .. ..	1,000	0.5	3,612	1,990
Coal gas .. ..	560	6	3,710	2,045
" " " " .. ..	475	10	3,710	2,045
Blue water gas .. ..	295	9	3,780	2,080
Hydrogen .. ..	320	nil	3,960	2,180
Carbon monoxide .. ..	318	nil	3,960	2,180
Producer gas .. ..	165	56	3,280	1,800
" " " " .. ..	128	61	3,050	1,690
Blast furnace gas .. ..	92	71	2,660	1,460

The lower flame temperature of the last three gases is due to their high content of inerts. The flame temperature can be raised by preheating both air and gas as indicated in Fig. 19. This fact also indicates the very considerable influence of excess air in cooling the flames.

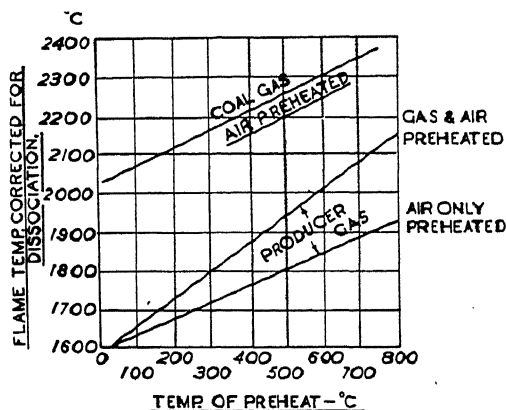


FIG. 19. The effect on flame temperature of preheating air.  
(A. Fells, *Trans. Inst. Gas E.*, 83, 612.)

## CHAPTER VI

## COMBUSTION IN PRACTICE

The art of securing proper combustion—Draught—The calculation of natural draught—Dimensions of the chimney—Primary and secondary air—Flow of air through fuel beds—Excess air—Flue gas losses—Time, temperature and turbulence as factors in combustion technique—Physical characteristics of the fuel bed—Combustion of various fuels and industrial combustion processes—Clinker formation—Firebed cooling—The utilisation of draught.

## THE ART OF SECURING PROPER COMBUSTION

**T**HE basic requirements for complete combustion are that the air and fuel shall be brought into contact under the right conditions and in the right proportions. To bring the air into contact with the fuel is the function of draught. Excess air above that needed to effect complete combustion causes waste of fuel as has been pointed out in Chapter V.

The conditions that must be fulfilled may be summarised as : an adequate and properly proportioned air supply introduced in the right place, together with adequate time, turbulence, and temperature to enable the combustion reactions to be effected and completed within the combustion space.

## THE AIR SUPPLY

It is necessary to bring the air into contact with the fuel and to remove the products of combustion when they are formed. To do this there must be a flow

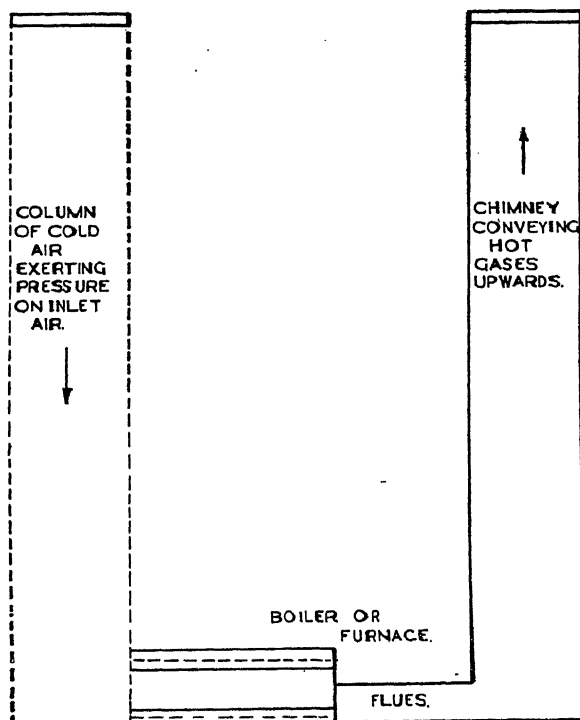


FIG. 20. Diagram illustrating natural draught and showing similarity between a chimney and a U-tube having fluids of different density in each limb.

of gases through the system as illustrated in Fig. 20. Air enters the system and passes through the furnace. Here it becomes heated by the combustion reactions in which it takes part. The gases formed in the furnace pass through the remainder of the plant to which they must give up the greater part of their heat and finally flow up the chimney. The flow of the gases is discussed in Chapter IX and the transmission of heat in Chapter VIII.

The movement of the gases, including the admission of air to the furnace, is effected in several ways :—

- (a) The air may be drawn in by the natural draught of a chimney, or
- (b) by an induced draught fan at the chimney base.
- (c) The air may be forced in under pressure by a fan, this pressure serving to propel the gases through the firebed, a chimney being used to exhaust the gases.
- (d) The air may be forced in under pressure by induction produced by a steam jet, a chimney being used as in (c).
- (e) The air may be forced in by a pressure fan and the products of combustion withdrawn by a suction fan in such a way as to leave zero pressure in the combustion space above the firebed, this being known as "balanced draught."

## DRAUGHT

Inefficient combustion is frequently due to the draught being inadequate to burn the amount of fuel fed on to the grate, or to failure to make use of the draught available. Deficiencies in this respect will also affect the output of steam from a boiler plant.

Neglecting at this stage the resistance of the firebed and other resistances such as may arise from dirty flues, etc., the amount of air that can be drawn into the furnace is governed by the cross-sectional areas of the flues and chimney and by the amount of draught. Since the dimensions are fixed in an existing plant only the draught can be varied. Natural draught is governed by (a) the height of the chimney and (b) the average temperature of the gases in the chimney. The higher the chimney and the hotter the gases, the greater is the draught. As methods of obtaining increased draught, both are expensive, the one owing to capital costs and the other because to create draught heat has to be used that could be converted into steam or used in other ways.

Thus if the provision of adequate draught necessitates an outlet gas temperature  $100^{\circ}\text{F.}$  higher than it need otherwise be, the heat used to produce draught per pound of coal burnt (see Table 29) would be

$$220 \times 0.0195 \times 100 = 430 \text{ B.Th.U.} \\ \text{cu. ft. of gas} \times \text{sp. ht.} \times \text{temp. rise}$$

or about 3 per cent. of the heat of the coal. In monetary terms this means 3 per cent. of the total cost of coal and handling charges.

Mechanical draught may be increased by installing larger fans. When putting in new boilers, ample margin should be allowed for possible overloads; many difficulties are due to lack of this precaution.

The use of steam jets requires caution. Normally the steam used should not be more than 5 per cent. of the total output of the boiler, but this is often much exceeded because it is not realised that erosion by the steam passing through the jet will wear away the metal. The steam discharged through a jet is given by the formula

$$\text{Steam passed (lb. per hour)} = 51 A(P + 15)$$

where  $A$  is the area of the jet in square inches and  $P$  is the gauge pressure of the steam in lb. per square inch. Since the area of the jet varies as the

square of the diameter, a very small amount of wear involves a disproportionately great amount of steam used. Fine jets operated at high pressure are more economical in their induction effect in introducing air than large jets with low pressure.

Table 28, showing steam discharged from jets, has been calculated from the preceding formula.

TABLE 28

Gauge pressure lb./sq. in.	Lb. of sat. steam passed/hr. by each jet having diameter :—			
	$\frac{1}{32}$ in.	$\frac{1}{16}$ in.	$\frac{1}{8}$ in.	$\frac{1}{4}$ in.
10	1	4	16	63
35	2	8	32	126
60	3	12	47	189
85	4	16	63	252

There are advantages in balanced draught, the chief being that air does not leak into the furnace through cracks or imperfections in the boiler front, neither is air drawn in nor flames blown out when the boiler or furnace door is opened.

Among the advantages of mechanical draught are :—

- (1) Cheaper low-grade fuels can be used, and the rate of evaporation increased by the additional draught available.
- (2) Combustion is more efficient and smoke is largely prevented by the provision of adequate air supply.
- (3) Furnace control is simplified.
- (4) Lower chimneys can be used.

Mechanical draught is discussed in Chapter XIV; in this present chapter attention is concentrated on natural draught.

## CALCULATION OF NATURAL DRAUGHT

The limitations of the available draught will be made clearer by indicating the basic cause of air movement.

The chimney, the furnace and the outer air form in effect a gigantic U-tube, (Fig. 20). The chimney is one vertical limb; the other equal and similar limb is a hypothetical column of cold air, and the furnace or boiler with the inter-connecting flues forms the lower, horizontal portion of the U-tube. For all practical purposes the density of air and flue gas are equal, and will here be taken as 0.076 lb. per cubic foot at 60° F. at atmospheric pressure.\*

If the chimney is 100 feet high, the weight of a column of cold air at 60° F. 1 square foot in section and 100 feet high is 7.6 lb.

Taking the average temperature in the chimney as 350° F., the weight of a column of waste gases of the same dimensions is inversely as the relative volumes at 60° F. and 350° F. and (cf. Chapter IV) is given by :—

\* This is virtually true for high-volatile coals burnt with 50–100 per cent. excess air. For coal and coke generally the figure is 0.077–0.082, being higher for the lower-volatile fuels, and lower as the amount of excess air increases. For town gas the density of flue gas is generally between 0.073 and 0.075. These figures refer to the gas containing its full content of water vapour as leaving the combustion chamber but calculated to 60° F. and assuming that the water is not condensed.

Since the contents of the limb of the U-tube containing atmospheric air are heavier (and are continuously maintained heavier through the operation of the furnace) than the contents of the other limb, a continuous motion will be set up, causing the heavier atmospheric air to descend and the lighter hot flue gas in the chimney limb to rise.

The motive power is thus a pressure of  $7.6 - 4.9$  or  $2.7$  lb. per square foot. As a column of water 1 square foot area and 1 inch high weighs  $5.2$  lb., the static draught is thus calculated as being  $(2.7/5.2 =)$   $0.52$  inches w.g. The measured draught with the gases moving up the chimney will be less than

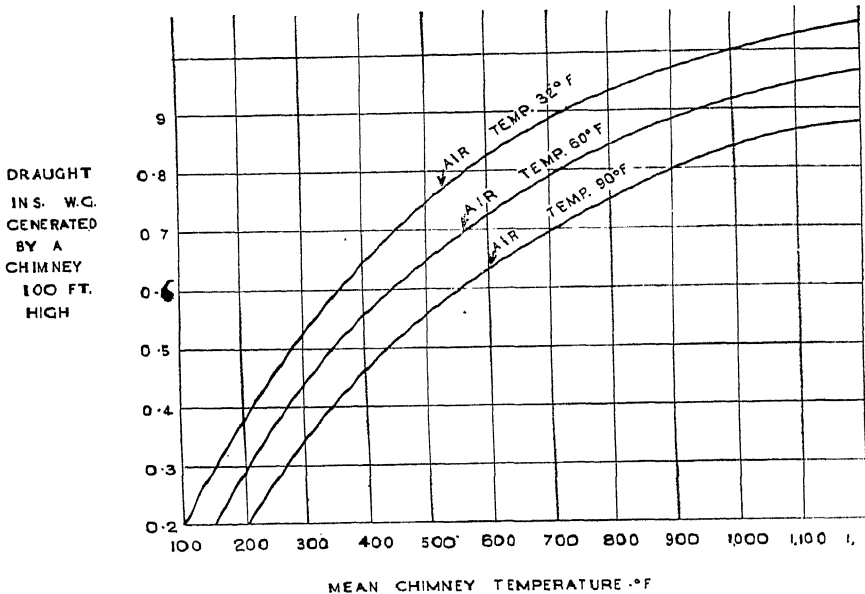


Fig. 21. Effects of gas and air temperature on static draught.

this since movement converts part of the pressure energy into energy of motion, as is described at the end of this chapter.

The following table gives the results of a calculation on similar lines for a chimney 100 feet high (a) with various average chimney temperatures and (b) with air temperatures representing a cold day and a hot summer day. The effect on the draught of these various conditions is striking (cf. Fig. 21).

	(a)			(b)	
	60	350	500	32	90
Outside air temp.—°F .. ..	60	350	500	32	90
Mean chimney temp.—°F. .. ..	200	350	500	350	350
Draught— <i>inches w.g.</i> .. ..	0.31	0.52	0.67	0.59	0.44

The chimney height depends on the draught required, and may also be calculated from the formula:—

$$7.6 - 7.9 \\ 460 - t_2 + 460.$$

## COMBUSTION IN PRACTICE

where  $H$  = height of chimney in feet.

$D$  = draught in inches w.g.

$t_1$  = outer air temperature, e.g.  $60^\circ\text{F}$ .

$t_2$  = mean temperature in the chimney in  $^\circ\text{F}$ .

7.6 & 7.9 = weights in lb. of 100 cubic feet of air and flue gas at  $60^\circ\text{F}$ .

For a chimney of given height and having a known mean temperature this formula can also be used to ascertain the draught that should be available.

The draught required depends on the conditions as is described at the end of this chapter.

The mean temperature in the chimney can be deduced from the temperature at the chimney base by subtracting  $\frac{1}{2}^\circ\text{F}$ . from the recorded chimney base temperature for every 3 feet of height.

The gas velocity that can be set up in a system of flues varies as the square root of the draught. For practical purposes the area of a chimney should be computed on the basis of a flue and chimney velocity of 15 feet per second measured at the working temperature. The gross area of chimney and flues are calculated on this basis from the probable quantity of gaseous products of combustion deduced from Table 15 allowing the proper quantity of excess air for the fuel concerned.

To the radius of the chimney as thus determined an additional 2 inches is added to allow for the diminished velocity against the walls.

### PRIMARY AND SECONDARY AIR

Air admitted directly to or with the fuel is known as primary air. With solid fuels primary air passes through the fuel bed. With gas, oil and powdered coal it is admitted through the burner and therein mixed with the fuel before combustion.

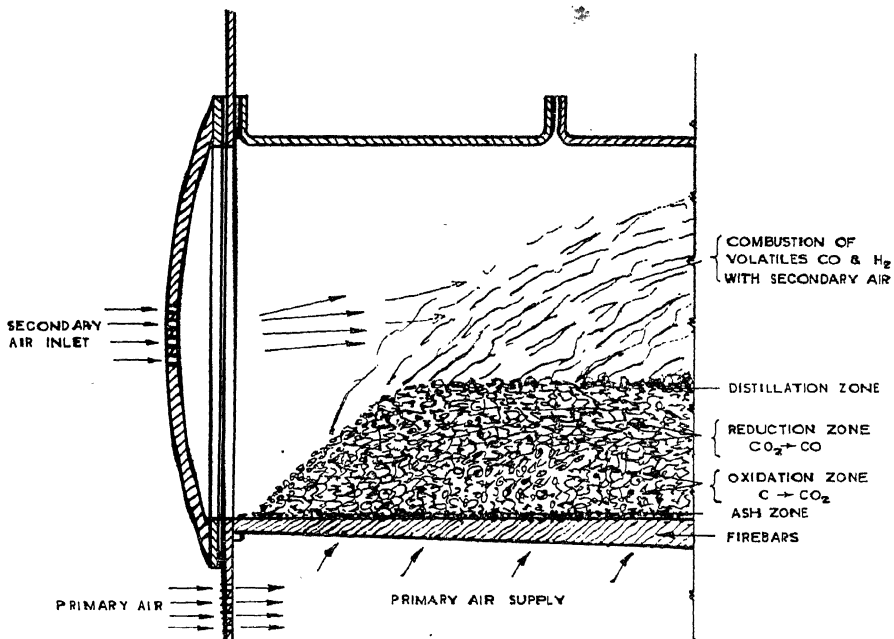
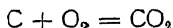


FIG. 22. Firebed zones in over-feed firing.

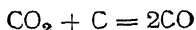


For gaseous and liquid fuels and powdered coal the remainder of the air necessary for combustion is admitted at a port placed (a) alongside the fuel burner or (b) farther along the furnace; for solid fuels which evolve combustible volatile matter a further air supply must be admitted above the fuel bed. This is known as secondary air. Some secondary air may find its way up the side or back of the fuel bed or even through thin places in the fuel bed. Combustion thus takes place in two stages. A third stage, tertiary air, is occasionally used.

The firebed may be divided into zones, as in Fig. 22. At the bottom lies the bed of ash. Above this is the oxidation zone in which the reaction between carbon and the oxygen of the air forms carbon dioxide directly:—



If the firebed is deep enough, the carbon dioxide is reduced by passage over further hot carbon forming carbon monoxide:—



Since carbon monoxide is a combustible gas, secondary air must be provided to burn it. In a shallow fuel bed very little carbon monoxide is formed.

Above this is the distillation zone. Raw coal thrown on to the hot fuel bed is carbonised or distilled, giving off its tarry gases into the combustion space above the fuel bed, where they must be burnt with the secondary air. For low-volatile fuels such as coke there is no distillation zone.

Combustion is thus brought about by the use of both primary and secondary air, and efficient combustion depends on the correct proportioning of the two. The secondary air must be so controlled and mixed with the gases leaving the fuel bed as to complete combustion within the appointed space and with as little excess air as possible.

If the fuel bed is sufficiently thin, the total air may sometimes be admitted by passing greater quantities of air through the bed than can react with the fuel in the time available. The attainment of the correct quantity of air is then not too difficult, but in induced draught installations where the secondary air is obtained by judicious opening of the air slides, or

TABLE 21A

	Bituminous coal	Coke
Ash .. .. .	7 per cent.	7 per cent.
"Fixed carbon" (coke residue) .. ..	63 " "	90 " "
Volatile matter .. .. .	30	3
Comp. of V.M. per lb. fuel:—		
CO and H <sub>2</sub> .. .. .	—	0.03 lb.
Water .. .. .	0.06 lb.	—
Tar .. .. .	0.06 "	—
Gas (taken as coal gas) .. .. .	0.18 "	—
Secondary air:—		
Air required to burn tar in the volatile matter at 154 cu. ft. per lb. .. ..	9.2 cu. ft.	—
Air required to burn distillation gases at 172 cu. ft. per lb. .. ..	31.0 cu. ft.	—
Air required to burn CO + H <sub>2</sub> .. ..	—	4 cu. ft.
	40.2 cu. ft.	4 cu. ft.

often from adventitious leakage round the grate, the fireman is faced with a far more difficult task.

Generally, too much excess secondary air is derived from leakages and the most that the fireman can do is to see that these are reduced and that his fire is in good condition. The effect of leakage is most pronounced when the burning rate per square foot is low. One of the advantages of a high burning rate is that the volume of combustible gases released is much greater, and is therefore better able to utilise the excess air arising from leakages around the firing zone and grate.

The calculation in Table 21A indicates the importance of correctly adjusting the primary and secondary air streams where the system of air admission requires that this should be done. For this purpose two dissimilar dry solid fuels are considered, taken from Table 15 (p. 54).

This calculation thus shows that with a low-volatile solid fuel such as coke, anthracite, steam coal, etc., by far the greater proportion of the air must be primary; with the higher-volatile coals, a great deal of secondary air must be admitted to burn the volatile matter. This calculation does not complete the picture, because the whole of the excess air—say, 50 per cent. for the coal, and 25 per cent. for the coke—will be admitted as secondary air; the complete picture is thus:—

	Bituminous coal	Coke
Primary air .. ..	94.6 cu. ft.	135 cu. ft.
Secondary air:—		
Air for combustion .. 40.2	4.0	
Excess air .. .. 67.4	34.7	
Total secondary air ..	107.6 cu. ft.	38.7 cu. ft.
Total air per 1 lb. dry fuel .. ..	202.2 cu. ft.	173.7 cu. ft.

A secondary air inlet correctly adjusted will offer a resistance to flow of air of the same order as that of the fuel bed. But whereas the fuel bed resistance will be approximately constant so long as the fire is well maintained, clean and without holes, as the air slide is opened its resistance decreases very rapidly. Unless the secondary air slides are properly adjusted, therefore, a given suction above the firebed will thus tend to draw in air through the secondary air openings rather than through the fuel bed. Some consideration of the resistance of a fuel bed to the flow of gases will be of interest.

## FLOW OF AIR THROUGH FUEL BEDS

The flow of air through a fuel bed is complicated because both streamline and turbulent flow can exist side by side. In streamline flow which occurs at low velocities, the pressure required to overcome the resistance of the fuel bed is directly proportional to the velocity (cf. Chapter IX). In turbulent flow the pressure loss is directly proportional to the square of the velocity,  $v$ ,

i.e. for streamline flow:  $p$  varies as  $v$   
for turbulent flow:  $p$  varies as  $v^2$

In a fuel bed the passages for the flow of air are dependent upon the **size** of the fuel and the character of the changes which are occurring, which it will be appreciated, must vary in turn with change of velocity. Further, the combustion of the air in being converted to carbon monoxide or carbon dioxide and steam takes place with a change of volume according to the proportions

of these various gases being produced. These vary throughout the fuel bed according to conditions. Again the volume of the gases is affected by the temperature in the fuel, so that the whole course of the flow becomes complicated and cannot be related to any simple statement of principles.

There is experimental evidence that the flow of air through a fuel bed is partly streamline, but is primarily turbulent, and that

$$p \text{ varies as } v^{1.72}$$

For practical purposes it can be taken that the flow is substantially turbulent. This means that to double the rate of combustion (which involves doubling the air supplied to the furnace) the draught must be quadrupled, the "draught" here meaning the pressure difference between top and bottom of the fuel bed.

## EXCESS AIR

The relative ease with which secondary air is drawn into a furnace directs attention to the importance of control of the quantity of excess air used for combustion. The methods of calculation of flue gas analysis given in Chapter IV enable the amount of excess air used and the total quantity of flue gases produced to be deduced from the flue gas analysis. The exact figures will depend on the analysis of the fuel, but guidance can be obtained from Table 29.

TABLE 29. EXCESS AIR, CO<sub>2</sub> IN FLUE GASES AND TOTAL VOLUME OF GASES

Excess air per cent.	CO <sub>2</sub> content of flue gases—per cent.			Total volume of gases (approximate) cu. ft./lb. coal (somewhat less for coke and anthracite)
	Bituminous coal 40 per cent. V.M.	Steam coal 11 per cent. V.M.	Coke	
0	18.9	19.6	20.5	150
30	14.6	14.9	15.8	200
50	12.5	12.9	13.6	220
100	9.4	9.7	10.2	300
150	7.5	7.7	8.2	375
200	6.2	6.4	6.8	450
300	4.7	4.8	5.1	600

The presence of unburned gas, e.g. carbon monoxide, in the flue gases is an important source of heat loss should it be allowed to occur. In boiler practice good results are obtained when the percentage of CO<sub>2</sub> is between 10 and 14 per cent. using coal, the figure depending on the character of the fuel and the type of equipment.

The proportion of excess air necessary to give complete combustion depends on the temperature in the combustion chamber. As a general rule, with anything over 14 per cent. CO<sub>2</sub> it is likely that unburned gases such as carbon monoxide and hydrogen will be present, which will cause a loss of efficiency greater than anything gained by a higher percentage of CO<sub>2</sub>. The presence of 1 per cent. of CO in the gases causes a loss of heat equivalent to between 4 and 5 per cent. of the coal burned; 0.1 per cent. of CO is equivalent to a heat loss of 0.47 per cent.; 0.1 per cent. of hydrogen is equivalent to a heat loss of 0.48 per cent.; 0.1 per cent. of methane (CH<sub>4</sub>) is equivalent to a heat loss of 1.52 per cent. For the efficient working of Lancashire boilers and similar types of installation, the CO<sub>2</sub> in the flue gases should generally be in the neighbourhood of 12 per cent.

Whilst a certain amount of excess air is for practical reasons unavoidable,

the waste resulting from too much excess air must be emphasised. The heat carried away in the waste gases depends upon their heat capacity which in turn depends mainly on their quantity and temperature. The whole of the useful heat cannot be abstracted by a boiler and its auxiliaries for example; economisers or air heaters may cool the gases to about 300° F.—generally to between 250° and 400° F. Accordingly if the volume of the waste gases is increased by excess air, so is the irreducible chimney loss for a given rate of combustion of coal.

For furnace work, as will be shown in Chapter XVIII, it is often necessary to ensure a reducing atmosphere within the furnace. For this purpose insufficient air is added for complete combustion, and the amount is adjusted to retain a little (up to 2 per cent.) of carbon monoxide in the flue gases, which is accompanied, when burning coal gas, by 2–4 per cent. of hydrogen.

The quantity of excess air that it is essential to use when firing with coal is generally not more than 50 per cent., and may be as low as 30 per cent. under specially favourable conditions. Coal gas and producer gas can be used with less than 10 per cent. of excess air, oil fuel with less than 15 per cent. and powdered fuel with less than 20 per cent., the essential factor being the uniformity with which fuel and air are fed into the combustion chamber. With gaseous fuel and almost to the same extent with liquid fuel, regulated streams of fuel are admitted at a uniform rate. This uniformity can be secured with coal to a larger extent by means of mechanical stokers and especially by firing powdered fuel; it cannot be secured with hand firing. Thus the amount of excess air admitted to the combustion chamber must depend upon the method of firing. Not infrequently 100 per cent. or more excess air is found to be used with hand fired boilers.

## GRAPHICAL CALCULATION OF HEAT LOSSES IN FLUE GAS

Detailed methods of calculating flue gas losses have been given in Chapter V. To simplify these methods in practice charts may be constructed similar to Figs. 14 and 15, and the heat content of the flue gases may then be directly read off the charts when the  $\text{CO}_2$  content of the gases (or the amount of excess air used) and the flue gas temperatures are known. Since the maximum  $\text{CO}_2$  content varies for different types of fuel, depending on the carbon and hydrogen content, charts must be calculated for the particular fuel used.

To avoid the labour of calculating complete sets of curves for each fuel, the more comprehensive method of Figs. 23 and 24 can be used.

Fig. 23 as here reproduced is drawn for four fuels, namely pure carbon, a bituminous coal, a fuel oil and a coal gas. It lends itself, however, to other compositions of the same fuels. It cannot be used for two-stage gases such as producer gas and blast furnace gas, for which the more elaborate curves previously described must be drawn, or the calculations made from first principles (Chapter V).

To construct Fig. 23, the straight line diagonally across the chart from the point (0, 0) to the point (21, 21) is taken as the basis. This line represents the theoretical  $\text{CO}_2$  content of the flue gas from any fuel when no excess air is present. On this line a point is marked corresponding to the theoretical  $\text{CO}_2$  content for the fuel used; thus for the fuels illustrated in the chart the points selected are:—

	Per cent. $\text{CO}_2$
Coke oven gas and town gas	11.5
Fuel oil .. ..	15.5
Bituminous coal .. ..	18.5
Pure carbon .. ..	21.0



If coke were used the point 20.5 would be selected. Some town and coke oven gases have a theoretical flue gas  $\text{CO}_2$  figure of 10.5 per cent. and some may be as high as 13 per cent. Whatever the figure may be for the fuel used, the position of the maximum  $\text{CO}_2$  content by analysis (i.e. with the water vapour condensed out of the gas) is marked on the basic diagonal line. This point is then joined to the point (0, 21) in the bottom right-hand corner of the chart, as has been done for the four fuels taken as typical.

The method of using Figs. 23 and 24 will be best explained by an example. Let it be assumed that oil fuel is used of such a composition that the theoretical flue gas contains 15.5 per cent.  $\text{CO}_2$  by analysis and that the flue gas temperature at the plant outlet is  $572^\circ \text{F.}$  ( $300^\circ \text{C.}$ ). The actual  $\text{CO}_2$  content of the gases is 10 per cent. From Fig. 23 it is found by measuring horizontally from 10 per cent.  $\text{CO}_2$  on the extreme left-hand side that with 10 per cent.  $\text{CO}_2$  the flue gases will contain 7.5 per cent.  $\text{O}_2$ . From Fig. 24, Curve 2, it is seen that 7.5 per cent.  $\text{O}_2$  indicates 55 per cent. excess combustion air. Then from the point where the waste gas temperature curve for  $300^\circ \text{C.}$  cuts the vertical line from 55 per cent. excess air, and reading horizontally on the extreme right, it is found that the stack loss is 16.5 per cent. of the net C.V. of the oil.

If an allowance of 5 per cent. be made for radiation and similar losses, the thermal efficiency of heat utilisation would be  $100 - 16.5 - 5 = 78.5$  per cent.

On the gross C.V. of the fuel the stack and radiation loss would be

$$\frac{(16.5 + 5) \times \text{net C.V.}}{\text{gross C.V.}}$$

## THE TIME FACTOR IN COMBUSTION

Having introduced the proper quantity of air into the furnace, divided when necessary in the right proportions of primary and secondary air, the further factors of time, temperature and turbulence may be considered.

All chemical reactions require a definite time for their completion, this being due either to the inherent nature of the reaction or to the difficulty of bringing the reactants in contact sufficiently quickly. Consequently a certain time must be allowed for combustion both in a solid fuel bed and in the gaseous combustion of the volatile matter or of a gas. With solid fuel, a bed which is too thin (or thin in relation to the size of the fuel used) enables air to pass through unburnt because there is not adequate contact between air and carbon. In gaseous combustion the volume of the combustion space must be adjusted to the conditions. For rapid combustion the air and gas (or powdered coal in pulverised fuel firing) must be as intimately mixed as possible immediately upon entry into the furnace. If the mixing is not intimate the flame may be very long and may extend outside the furnace proper into the flues.

## TURBULENCE

Turbulent flow as opposed to streamline flow is the means whereby air and gas may be mixed rapidly. Parallel streams of gas and air will diffuse slowly into one another and cause slow combustion. If the flow is turbulent, however, the air and gas are thoroughly mixed in a small volume so that the opportunity for bringing the oxygen and gas into contact is increased.

In boiler furnaces the volatile matter is evolved from the coal and the secondary air may tend to pass over the top of the volatiles as shown in Fig. 25. When this happens turbulent flow may be induced by a jet of steam or air directed downwards at an angle. One such arrangement which makes use of preheated air for the purpose is shown in Fig. 26.

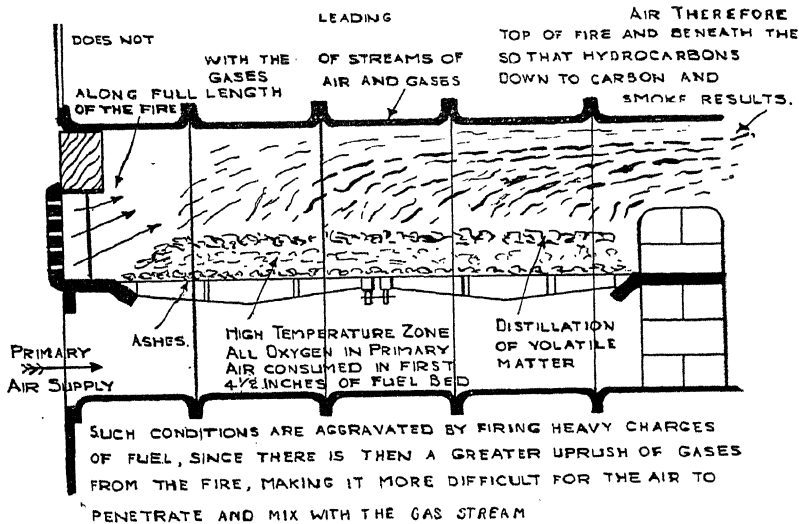


FIG. 25. Stratification effects in boiler furnaces.

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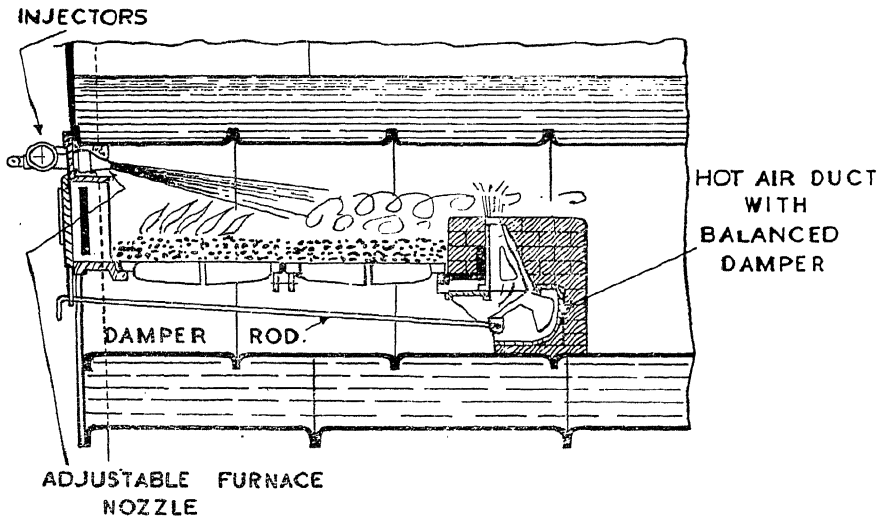


FIG. 26. A method of increasing turbulence in boiler furnaces.

## TEMPERATURE

Combustion will not take place unless the temperature is sufficiently high. The temperature of ignition of materials has been dealt with in a preceding section. Unless this temperature of ignition is exceeded combustion cannot take place. It is particularly necessary to mix the secondary air and the combustible gases as soon and as well as possible when there are cold surfaces in the proximity of the grate. It is in this respect that preheated secondary air has an advantage. This early mixing must be avoided in certain furnaces and kilns where long flames are required.

The speed of reaction increases and is commonly doubled or trebled for each  $10^{\circ}\text{C}$ . rise in temperature. Consequently a high temperature, which may be promoted by preheating air for example, will assist in obtaining more rapid combustion. It has been computed that at furnace temperatures, hydrocarbons whether contained in coal gas or the volatile matter from coal are decomposed completely into carbon and hydrogen in one-thirtieth of a second.

If production of black smoke is to be avoided from liquid fuels or from the volatile matter of coal this carbon must be burnt. An important influence upon this is the chilling effect of cold surfaces. The flames from burning coal are approximately at  $2,000^{\circ}\text{F}$ . ( $1,094^{\circ}\text{C}$ .) or higher according to the temperature in the fuel bed and the distance the flame has travelled from the hot incandescent fuel, whereas boiler metal, for example, is no higher than the saturated steam temperature, generally below  $450^{\circ}\text{F}$ . ( $230^{\circ}\text{C}$ .). In a furnace the stock will generally be charged cold. If the flames impinge on these cool surfaces combustion may be arrested and carbon monoxide and hydrogen may go forward into the flue gases. In extreme conditions carbon will remain unburned and will form smoke.

The chilling effect of cold surfaces on flames may go farther than this. If a basin of cold water is held in a luminous flame of coal gas the basin will quickly be covered with a film of carbon. In the same way soot may be deposited on metal surfaces. This carbon represents loss of fuel, as has just been indicated, and in addition it has a low thermal conductivity so that the heat transmission from the flame to the vessel being heated is reduced.

Chilling of furnace flames can arise from :—

- (a) Heavy infrequent charges.
- (b) Excessive air supply, e.g. by prolonged opening of fire doors.
- (c) A long flame which impinges on the metal, e.g. a flame from very high volatile coals.
- (d) Thermal features which force the burning gases into intimate contact with cooler metals, e.g. too high a fire-bridge in boilers.
- (e) The design of the plant, e.g. the circular section of shell boiler furnaces makes them more liable to this type of chilling than other types of boiler furnace.

## PHYSICAL CHARACTERISTICS OF THE FUEL BED

The importance of the condition of the fuel bed has been emphasised when dealing with the flow of air. Above the fuel bed there is a lower pressure than exists below the bed. This pressure difference, aided by the buoyancy of the hot gases within the fuel bed, causes air to flow through it. The amount of fuel burnt depends upon the amount of air that flows through the bed, provided that the air interacts with the fuel (cf. Table 30).

Clearly the ideal conditions are that

- (1) the fuel bed shall offer uniform resistance to the flow of gases at all parts where there is combustible fuel to be burnt,



- (2) the fuel shall be of such a size that it will react rapidly and uniformly with the incoming air.

The practical characteristics of the fuel bed will now be discussed.

*Size of Coal.* In order to make the fullest use of primary air it is essential to have a uniform fuel bed. A small-sized, graded fuel gives the best air distribution. A small percentage of fines through  $\frac{1}{8}$  inch mesh may not be a disadvantage, but the quantities of fines found in slack increase the fuel bed, resistance and often cause uneven air distribution. This is largely due to segregation, i.e. separation of larger material from the fines. Some parts of the fuel bed will therefore burn out more quickly than others, leaving bare patches, through which unburned primary air rushes uncontrolled.

This difficulty can be obviated to a great extent, especially in chain-grate stokers, by thoroughly wetting the coal. The amount of water to be added depends on the percentage of fines under  $\frac{1}{8}$  inch mesh. For a normal slack containing 20–40 per cent. fines, 5–7½ per cent. of water is required, rising to 12½ per cent. for fine slacks with 80 per cent. through  $\frac{1}{8}$  inch mesh. This water is in addition to the inherent moisture that remains in the coal when air-dried. For really fine coals the water content may require to be over 20 per cent.

It is essential for firing on chain-grate stokers that slack should be uniformly wetted. Since this is a difficult operation to perform in practice, there is an advantage in using washed slack, already evenly wetted, instead of dry slack. The addition of water results in a small lowering of the efficiency, to the extent of approximately 0·1 per cent. for each 1 per cent. of water added, but this is far more than fully recovered by improved combustion conditions. The objection to *buying* wet slack is that the water is paid for at the price of coal.

*Coking Properties.* The coking properties of fuel affect the physical character of a fuel bed to a marked degree. Plastic coal will not permit the passage of air and in addition a coking coal tends to give an open, irregular fuel bed, made up of large masses of coke separated by wide fissures, which give bad contact between air and fuel. As a result, a greater depth is required for complete primary combustion and the amount of  $\text{CO}_2$  reduced to CO is small.

A non-caking, free-burning coal, on the other hand, gives a uniform fuel bed, made up of particles in close contact, so that both primary combustion and secondary reduction proceed rapidly and efficiently. Under certain conditions, fuels may form a crust on top of the bed and stop air penetration; this crust must be carefully broken up.

*Thickness of Fire.* A thick fuel bed offers more resistance to the flow of air than a thin one, and therefore with the same available draught less primary air passes through a thick bed. If difficulty is experienced in maintaining steam in a boiler with a thick fuel bed, it may be possible to overcome the difficulty by carrying a thinner one, but it is, of course, rather more difficult to avoid holes in a thin bed. Some mechanical stokers operate with a thin fuel bed (cf. Chapter XI).

With bituminous coals, the firebed thickness is generally from 3 to 5 inches with chain-grate stokers and up to 8 inches with hand firing.

## THE COMBUSTION OF SOUTH WALES SLACK

South Wales low-volatile slack is difficult to burn with natural draught, but burns satisfactorily with forced draught. The coal has a high calorific value, on the dry ash-free basis, of the order of 15,500–15,650 B.Th.U. per lb. as compared with 13,850 for low rank Midland coals, and it also has a low inherent moisture content. On the same ash basis, therefore, the weight of South Wales coal which must be burned to give the same heat release is three-quarters of that of the low rank coal.

The same amount of total air is required per unit of heat generated, but a larger proportion of this must be primary air because the volatile content of the coal is lower ; moreover, the reactivity of the fuel is less with the net result that the gases leaving the fuel bed contain less CO and hydrogen, and therefore require less secondary air to complete combustion.

### THE COMBUSTION OF COKE

In the preceding sections, the effect of the most important factors governing the combustion of coal, its size, reactivity and volatile content and the depth of fuel bed, was considered. Coke yields so little volatile matter that the influence of volatiles on combustion can be neglected, but the other factors are as important as in the combustion of coal.

*Depth of Fuel Bed.* Table 30 shows the composition of the gas at different levels in a coke fuel bed. It will be seen from the figures that it is possible to distinguish the same two zones of combustion, as those described in the coal fuel bed.

CO<sub>2</sub> is formed in the bottom zone. Higher up, when all the oxygen is used it is reduced to CO. For example, with the  $1\frac{1}{2} \times \frac{1}{2}$  inch fuel, the oxygen is practically all used up  $4\frac{1}{2}$  inches from the bottom of the bed, and increasing quantities of CO are being formed.

At the top of the bed, almost all the CO<sub>2</sub> has been reduced and the larger quantities of CO in the gas require secondary air to complete combustion.

*Size of Fuel.* The effect of size on combustion can be judged from the table by comparing the combustion of the two sizes of coke,  $3\frac{1}{2} \times 3$  inches and  $1\frac{1}{2} \times 1$  inch. With the larger size, there is less intimate contact between air and coke, and the combustion zone is therefore much deeper. At a level  $4\frac{1}{2}$  inches from the bottom less CO<sub>2</sub> has been formed, and the oxygen has not been used up to the same extent. Even at 6 inches, little CO has been formed.

At the top of the bed, there is much more CO<sub>2</sub> and much less CO than with the smaller size, so that less secondary air will be required to complete combustion. The lack of contact between the air and the fuel results in a much lower burning rate for large coke than for the smaller size.

TABLE 30. GAS COMPOSITION IN A COKE FUEL BED

Distance from bottom of fuel bed	Composition by volume—per cent.					
	Size of coke $3\frac{1}{2} \times 3$ inches			Size of coke $1\frac{1}{2} \times 1$ inch.		
	CO <sub>2</sub>	O <sub>2</sub>	CO	CO <sub>2</sub>	O <sub>2</sub>	CO
12 inches .. ..	14.0	0.5	6.4	5.4	0.4	24.9
6 " " " " " "	16.0	4.0	0.5	15.1	1.1	7.0
$4\frac{1}{2}$ " " " " " "	12.0	7.5	1.0	16.3	1.4	5.3
3 " " " " " "	—	—	—	16.8	2.9	2.7
$1\frac{1}{2}$ " " " " " "	5.0	13.5	1.0	11.5	8.0	0.8

If the size of the pieces of fuel is sufficiently great, some air may pass through the bed without contact ; this is known as channelling. When this happens, the unused air functions as secondary air.

The smaller sizes of coke have the disadvantage of tending to form objectionable clinker. This is partly due to the fact that the ash is evenly distributed through the pieces of coke and partly to their small size. Both lead to close contact, which is a primary cause of clinker formation. Difficulty with clinker formation may be lessened by using large fuel. This is also true of anthracite.

*Reactivity and Combustibility of Coke.* A characteristic of coke is that different types react at different rates with oxygen and  $\text{CO}_2$ . The results recorded in Table 30 relate to oven coke. With gas coke the results might be somewhat different in that more rapid combustion in the lowest zone and more rapid reduction in the next zone would probably occur.

There are also variations in the reactivity of the coke formed during the combustion of coal, due to variations of conditions within the fire. Generally it seems that the coke formed at high rates of combustion is more reactive than that formed at low rates.

Coke burns more slowly than coal and requires more draught because a much greater proportion of primary air is needed. In boiler practice it is frequently found that sufficient secondary air is derived from air passed through or round the grate or entering through imperfections in the fitting of the doors, and that the regular secondary air slides should be closed. This applies also to anthracite. The composition of the combustion gases from coke differs somewhat from those from coal (see Table 29).

## SMOKE

Smoke is formed from the tarry distillation products of coal, which escape before combustion is complete. When the amount of volatile matter in the coal becomes small, the quantity of tar evolved from the coal becomes much less, and the less is the liability to smoke; this is the reason for the smokeless characteristics of Welsh smokeless coal, anthracite and coke.

There are two forms of smoke: (1) brown smoke, which is mostly tarry matter given off at relatively low temperatures when fresh coal is fired on to a hot surface and which is relatively easy to consume, and (2) black smoke, which consists largely of particles of carbon derived from the cracking of hydrocarbon gases at a high temperature, and is probably formed when the fuel bed is stirred. Excess of air even if preheated is not sufficient to burn this black smoke; it must also be subjected to a high temperature when in contact with air. The two smokes may, of course, occur simultaneously.

To consume smoke there must be sufficient air, adequate mixing of air and volatiles, and the temperature must be sufficiently high to enable combustion to take place rapidly and before the gases come into contact with cold surfaces.

The amount of carbon in dense smoke amounts to not more than 1 per cent. of the carbon in the coal; the real danger lies in the possible presence of other unconsumed combustible gases. On the other hand, attempts to avoid smoke by using considerable quantities of excess air may cause much greater losses. A chimney serving a furnace working on bituminous coal should be neither clear nor smoky; the gases should be just visible at the top and a wisp of brown smoke, in boiler practice for example, may be indicative of the most efficient conditions. A plant that never shows a trace of smoke is not burning coal efficiently.

## THE COMBUSTION PROCESS IN OVERFEED FIRING

The process of combustion varies, to some extent, from one form of appliance to another and the following examples give an indication of how variations in the firebed conditions affect the proportions of primary and secondary air required.

With a thick fuel bed (cf. Fig. 22), where the raw coal is fed on to the top of the hot fuel most of the volatile matter may be driven off before the carbonaceous residue descends to the lower levels of the bed where an excess of oxygen is present. Under such conditions the volatile matter distilled unburned requires a supply of secondary air over the fuel bed for its combustion.

With a thin fuel bed, of the order of 3 inches in depth as on a sprinkler

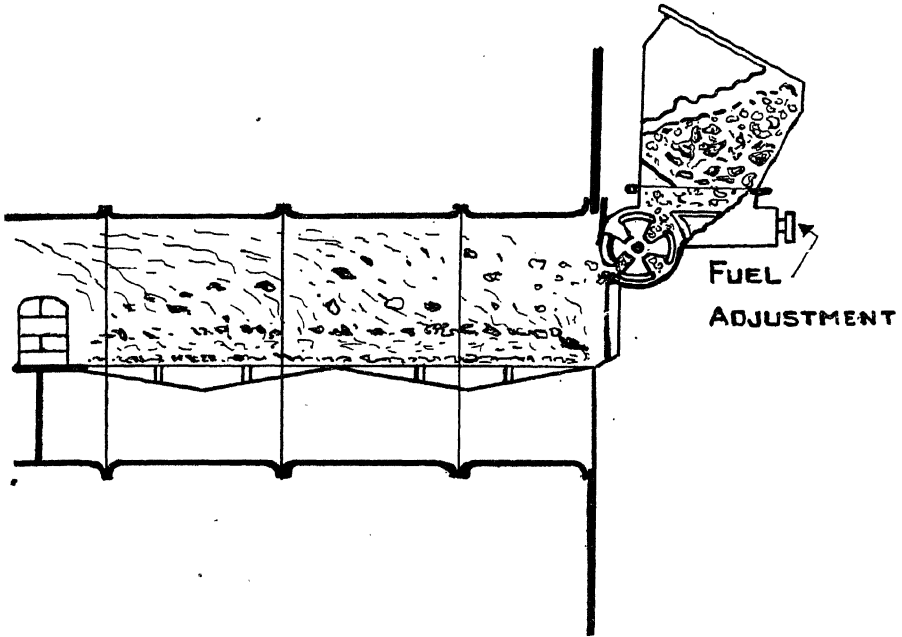


Fig. 27. Sprinkler stoker.

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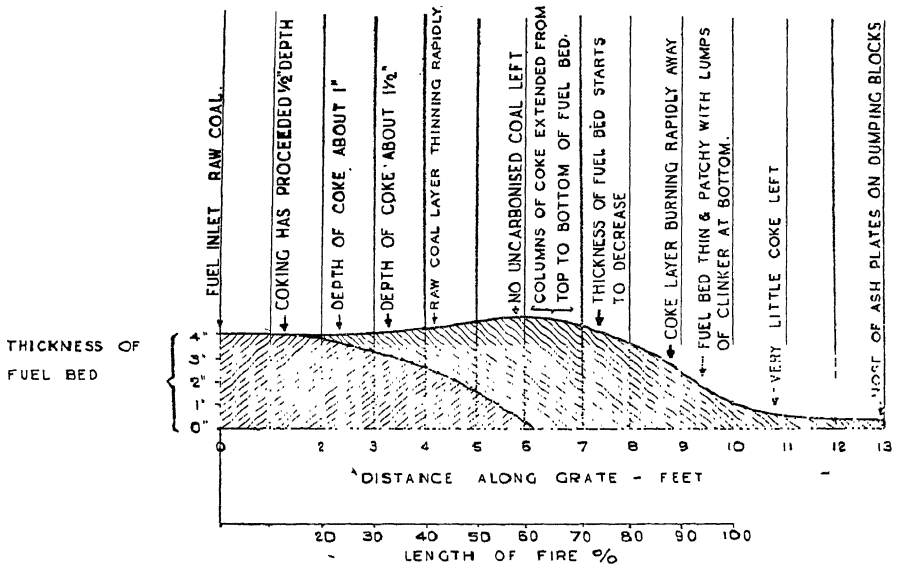


Fig. 28. Combustion on a chain-grate stoker  
(B. M. Thornton, *J. Inst. Fuel*, VIII, 13.)

stoker (Fig. 27), the reduction zone is non-existent, and distillation and combustion are brought together. A proportion of the volatile matter is burned by the primary air and much less secondary air is required.

With a travelling grate stoker (Fig. 28) very little volatile matter is given off unburned, except in the extreme front of the grate during the process of ignition. Most of it is burned just above the advancing ignition plane, although part of the  $\text{CO}_2$  is subsequently reduced to CO in the upper levels of the fuel bed.

### THE COMBUSTION PROCESS IN UNDERFEED FIRING

Fig. 29 illustrates the process of combustion in an underfeed stoker, where conditions are quite different from those of overfeed firing. The raw coal

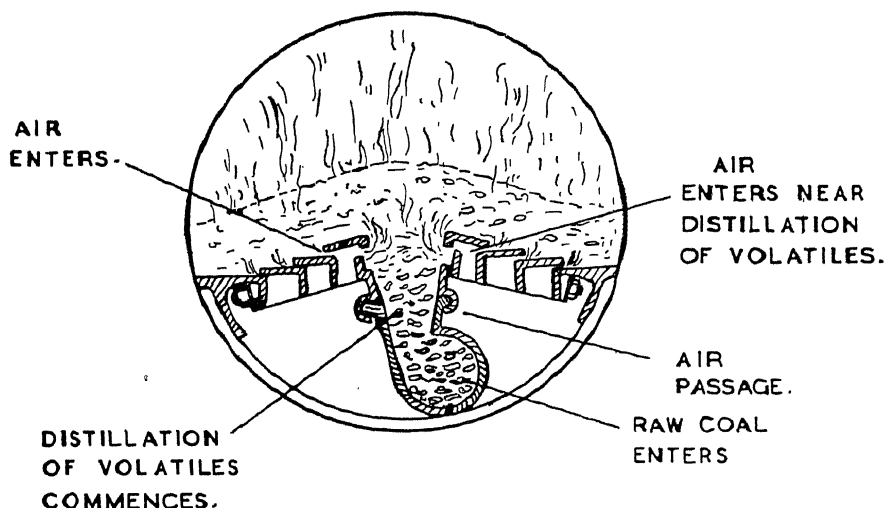


FIG. 29. Combustion in an underfeed stoker.

enters at the bottom of the firebed and is forced upwards, into a zone where air is supplied. The volatile matter is therefore released towards the bottom of the bed and is burnt in the hottest zone just above it.

### CLINKER FORMATION

One of the major difficulties in maintaining a uniform resistance of the fuel bed is the formation of clinker. Masses of clinker prevent the flow of air effectively and the portions of the grate which they cover are thus sealed, greater work being thrown on the rest of the grate. Since the resistance of the fuel bed varies as  $v^{1.72}$ , the draught required for a given combustion rate is considerably increased by the presence of clinker. A discussion of clinker and its formation is thus an essential part of the general subject of combustion.

The ash in coal consists partly of the inherent ash derived from the wood of the original coal forests, this ash being generally low in quantity but in a very finely divided condition. The greater part of the ash consists of shale and similar carbonaceous mineral matter associated with the coal. This portion of the ash may vary in size from powder up to quite large pieces. There may be an equally wide variation in the chemical composition of the several particles of ash.

An ash may thus consist of a mixture of comparatively fusible with infusible substances, and clinker formation may occur by the fusible substances binding

together the infusible. Iron oxide which under reducing conditions can form very fusible ferrous silicate, has probably the predominating influence in the formation of clinker from solid fuel (Chapter XXII).

The melting point of coal ash is determined by grinding the ash to powder and determining the melting point of the mixture. Unless this mixture gives a low value, it cannot be said to be a criterion of liability to clinker formation because it takes no account of the fact that a small proportion of fusible material may bind the whole together into masses of clinker.

As a general guidance, and based upon the determination of ash melting point on the powdered mixture, the following figures have been given :—

- Group 1. Melting point, 1,425°–1,710° C. Clinkering troubles absent.
- Group 2. Melting point, 1,200°–1,425° C. Clinker may form, but is not likely to be serious.
- Group 3. Melting point, 1,040°–1,200° C. Clinkering troubles excessive, unless adequate precautions are taken.

The ash melting points of most British coals lie between 1,050° and 1,350° C. although there are notable exceptions such as the ashes of durains, many of which melt at over 1,450° C. Some of the ash constituents will usually melt below the temperature of the fuel bed (1,300°–1,500° C.) and will thus afford a basis for clinker formation.

Table 31 gives the results of analyses of nearly 800 coals which, whilst not necessarily being completely representative, give a fair picture of the melting points of ashes of British coals. These melting points were determined in a reducing atmosphere.

TABLE 31

° C. ° F.	Number of coals with ash melting points between stated limits					Total No. of coals tested
	1,100 2,012	1,100–1,200 2,012–2,192	1,200–1,300 2,192–2,372	1,300–1,400 2,372–2,552	1,400 2,552	
SECTION 1						
English coals (excluding Durham) . . . .	10	298	288	71	17	684
Expressed as per cent.	1½	43½	42	10½	2½	—
SECTION 2						
Durham . . . .	—	6	5	14	15	40
Scotland . . . .	—	3	7	6	12	28
South Wales . . . .	—	5	4	19	7	35
Total of Section 2 . .	—	14	16	39	34	103
Expressed as percentage . . . .	—	13½	15½	38½	33	—

Whether a given ash will form clinker or not will depend upon several circumstances :—

(a) The distribution of the mineral matter in the coal.

Clinker may form through complete fusion of the ash, which will lead to large sizes of clinker ; or by the cementing together of pieces of shale which are themselves relatively infusible, in the manner previously described. It can be shown that particular fractions can be obtained from coals, by a combination of sizing and float-and-sink tests, which melt much below the average temperature. Fusible and infusible constituents are thus not blended in a uniform manner in the coal, but may be distributed differently in different parts of the seam and in different laminations in each lump of coal.

(b) The atmosphere to which the ash is exposed.

A coal ash will melt at a lower temperature in a reducing atmosphere than in an oxidising atmosphere. Experiments suggest that for coals having a fusion point above 1,200° C. the difference is 30°-50° C. Below this the difference is greater as will be seen from the following figures :—

Melting range °C.	Temperature difference between oxidising and reducing conditions			
		°C.		°F.
1,150-1,200	..	80	..	144
1,100-1,150	..	175	..	315
1,050-1,100	..	200	..	360

In ashes from boilers and furnaces the iron is almost completely reduced to the ferrous state so that the conditions are the worst possible in respect of clinker formation. The decrease in fusion temperature is due to the formation of complex ferrous silicates. A thin fuel bed is the more likely to produce an oxidised ash and so to reduce clinker formation.

(c) The extent to which the mineral matter becomes mixed with the fuel in the hottest zone of the fuel bed.

Disturbance of the fuel bed during combustion tends to raise mineral matter from the grate (where it is cooled by air) into the hottest part of the fire. Slicing and poking should be kept to a minimum. In under-feed stoking the fuel bed is stirred up thus assisting the formation of clinker, whereas on chain grates, where the fuel bed is moved bodily forward, disturbance is a minimum.

Allied to this is the contact between the several particles of coal. This will be greater as the coal decreases in size, the smaller the coal the greater the likelihood of clinker formation. It has been suggested that the very finely divided inherent ash may have a greater effect on clinker formation than the rough associated dirt.

In those zones where the coal is burning most actively the temperature will be the highest. Much of the ash will then be within the coal, and ash particles may not be so readily brought into contact with one another as in the last stage, when most of the carbon has disappeared; in this last stage, however, the combustion is less intense and the temperature will fall. This view is confirmed by published observations (*J. Inst. Fuel*, 14, 223) showing that both with coal and coke clinker formation occurs more than 4 inches above the grate, and as it sinks through the fuel bed, the clinker solidifies below the 4-inch level. This illustrates the next point also.

(d) The time to which the ash is subjected to any high temperature.

Chemical reactions may take place with the formation of mixtures of low melting point and unless there is sufficient time given these low fusible constituents may not be formed if they do not exist already in the coal.\*

(e) For clinker to form in any quantity the conditions must be such that a sufficient amount of slag shall be produced which is fluid enough to permit it either to run together in masses or to percolate through other masses of infusible mineral matter so as to bind them into aggregates.

(f) Preheated air will raise the fuel bed temperature and thus favour clinker formation.

Further information on clinker formation may be found in Chapter XXII, under "The Action of Slags."

## FIREBED COOLING

Steam is sometimes used under fire-grates (in addition to its use in forced draught injectors)

\* Some ashes will form mixtures of minimum melting point, in which the melting point is less than that of the individual constituents. Quite frequently, however, the melting point of the mixture will be above that of the individual constituents.

## COMBUSTION IN PRACTICE

- (a) To reduce the grate temperature and prevent material damage.
- (b) To reduce the firebed temperature to diminish or prevent clinker formation.

Jets of steam are blown into the fuel bed from below somewhat in the manner indicated in Fig. 30. This use of steam is quite distinct from the use of steam jets to promote draught or pressure and the steam acts wholly as a cooling medium.

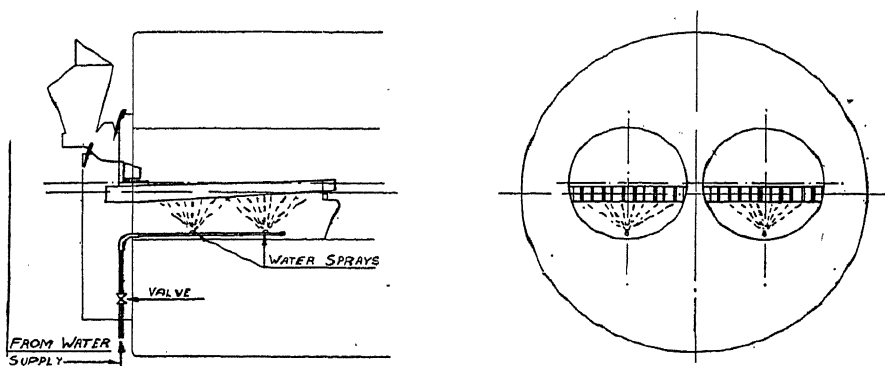


FIG. 30. Arrangements of water jets for cooling firebars.

The quantities of steam involved may, unknown to the operator, attain such dimensions as to cause a heavy strain on the available steam supplies. Published records suggest that 6 per cent. of the total steam production of the plant is not an unusual figure, though this is at least three times more than should be necessary. Wear on the nozzles and inattention whereby steam is left on unnecessarily, are responsible for much waste.

Steam used for cooling fire-grates operates in two ways :—

(1) Mechanical contact with the grate and the firebed results in the steam being heated to a high temperature and thus abstracting heat.

In considering this effect, three factors must be taken into account.

- (a) If saturated steam is contained in the primary air stream under the grate, the steam has a total heat of 1,150 B.Th.U. per lb. If in passing through the grate and lower portion of the firebed it is then superheated to  $1,000^{\circ}\text{C}$ . ( $1,832^{\circ}\text{F}$ .), its total heat becomes 1,994 B.Th.U., representing an abstraction of  $(1,994 - 1,150 =) 844$  B.Th.U. per lb. of steam used. Clearly it would be advantageous if water could be used instead of steam, thus permitting the additional 1,150 B.Th.U. per lb. to be applied for direct cooling, instead of representing coal wasted through additional generation of steam in the boiler.
- (b) This 844 B.Th.U. of heat is not necessarily used where it is most needed. The steam is raised in temperature primarily in the combustion zone, and is reasonably efficacious in cooling this part of the firebed. Its direct effect in cooling the grate is small. If water were used, on the other hand, a considerable proportion of the 1,150 B.Th.U. mentioned above would be available for directly cooling the grate or the clinker immediately above it.
- (c) Following the argument just developed in (b) it is instructive to observe the relative cooling effect of the primary combustion air and the steam or water admitted with it. A coal may require about 100 cubic feet of primary air per lb. of coal, and when cooling with steam it is customary,



as previously indicated, to use about 0.3 lb. steam per lb. of coal. If the air and steam are heated to 500° F. (260° C.) when passing through the grate and the ashes at the immediate base of the fire, it can be shown that :—

100 cubic feet of primary air absorbs	860 B.Th.U.
0.3 lb. steam absorbs	41     "
0.3 lb. water absorbs	386     "

The advantage of cooling firebars by water rather than steam is manifest from these considerations.

(2) The formation of water gas and consequent absorption of heat cools the firebed.

When steam is brought into contact with sufficiently hot carbon it forms the gases carbon monoxide, hydrogen and carbon dioxide and in so doing absorbs considerable quantities of heat.

The extent to which water gas is formed in a solid fuel furnace depends on the depth and temperature of the firebed, the size of the fuel, the reactivity of the fuel, and so forth. This reaction will not directly cool the grate, but helps to do so by reducing the temperature of the whole firebed.

According to the composition of the water gas, the thermal effect per lb. of steam decomposed will be an absorption of between 4,170 and 2,120 B.Th.U. per lb. of water or between 3,120 and 1,070 B.Th.U. per lb. of steam. Probably no more than half the steam will be decomposed, and if 0.3 lb. of steam are used per lb. of coal, and if the gases and undecomposed steam leave the fuel bed at 900° C. (1,652° F.), the total cooling effects per lb. of coal would then be :—

Water spray	between 1,104 and 774 B.Th.U. per lb. coal.
Steam admission	between 772 and 443     "     "     "

There is experimental evidence that the effect of saturating at 50°–60° C. the air blast of a gas producer with steam, involving the use of some 0.3–0.4 lb. steam per lb. of coal gasified, lowers the hot zone temperature by some 300° C. (540° F.), i.e. from 1,400° to 1,100° C. (2,550°–2,010° F.). It has been found (also experimentally) that in a travelling grate stoker it requires 1 per cent. of the steam generated in the boiler to lower the link temperature by 30° C. (53° F.) with a caking coal and 50° C. (90° F.) with a non-caking coal.

Danger to the firebars occurs principally after periods on high load. During high load periods the firebars are cooled by incoming primary air. The firebed meanwhile increases considerably in temperature. If a period of moderate load follows, the cooling effect of the incoming air is diminished, since its quantity is decreased, but the high firebed temperature increases the temperature of the grate. When a boiler is continuously on a moderate load it is generally possible to operate without artificially cooling the grates.

The considerations just advanced have led to investigations on the possibility of using water sprays instead of steam jets for cooling purposes. Fig. 30 shows the general arrangement and Fig. 31 the design of a nozzle that has been found satisfactory for the purpose (cf. B. M. Thornton, *Engineering*, September 4th, 1942, p. 142). It is essential that the spray of water should be very finely atomised, and provision should be made for draining any drip back from the firebars on to the flue.

The quantity of water required for cooling has been found to be only one-half to one-third that of the steam needed. Figures published by B. M. Thornton (*loc. cit.*) obtained on an Economic boiler of 15,000 lb. per hour are as follows :—

Gross steam generated per hour	15,000 lb.
Evaporation per lb. of coal	7.5 lb.

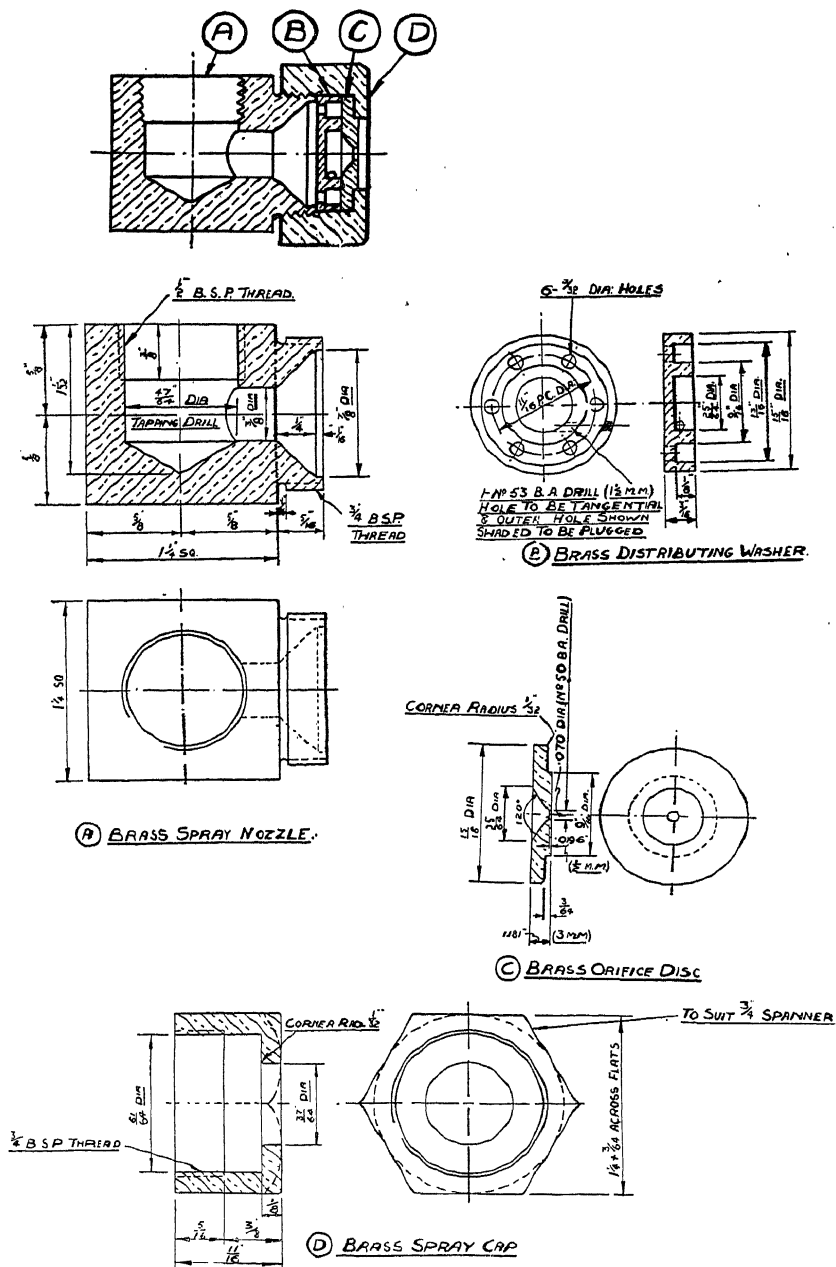


FIG. 31. Design of  $\frac{1}{2}$ -in. spray jet suitable for cooling firebars by water sprays (B. M. Thornton).

	Coal required per 1,000 lb. steam lb.
No cooling of firebed .. .. .	133
Cooling with 860 lb. steam per hour ..	140.5
Cooling with 600 lb. steam per hour ..	138.5
Cooling with water sprays, 270 lb. per hour	135

This discussion on the important subject of fire-grate cooling may be summarised thus :—

- (1) Cooling of the grate bars and firebed is often wasteful of fuel and should be severely restricted or avoided altogether wherever possible, but it may be found necessary with some types of solid-bar stoker and when dealing with an ash of low fusibility. Trials should be made without steam admission.
- (2) Important economies can be secured by cooling with atomised water jets in place of steam.
- (3) When cooling is necessary, the minimum quantity of cooling agent can be secured by fitting an orifice plate in the steam or water pipe. Trials should be made to find the minimum size of orifice that will give satisfactory results, starting with a  $\frac{1}{8}$ -inch hole (see Table 28). The maximum quantity of steam or water used will then be restricted though the amounts can be reduced. Since the orifice will wear it must be changed periodically. This minimum should be rigidly adhered to unless a change in the nature of the coal used compels an increase or makes possible a decrease.
- (4) The jets should be applied only to those portions of the grate where the heat is greatest. Areas of lower temperature, or where clinker does not form, should not be subjected to cooling.

## UTILISATION OF DRAUGHT

Thus far it has been presupposed that the plant is in good order and that there is no difficulty in getting all the draught required. It frequently happens, however, that industrial plants are short of draught and thus cannot maintain temperatures or output. Often the problem resolves itself into making proper use of whatever draught is available. The considerations that govern this will now be discussed in some detail. The effective utilisation of draught is one of the most important branches of the work of the combustion engineer.

The proper dimensions of the chimney can be calculated by the method already described. The draught available with a given height of chimney for a given temperature of the outside air is determined by the mean temperature in the chimney and this in turn depends on the gas temperature at the chimney base. Fig. 21 indicates this relationship for air temperatures of 32° F., 60° F. and 90° F. The effect of the gas temperature towards the lower end of the scale is more considerable than at higher temperatures. It is important when draught is supplied by a chimney to ascertain that there is no serious temperature drop between the outlet of the boiler plant or furnace and the chimney. Chimney flues when long are sometimes insulated with good results.

A drop in temperature along the chimney flue may be due to inleakage of cold air. This will be more serious than cooling alone, because it also increases the volume of gases to be evacuated by the chimney.

The volume of flue gases is an important factor, and this reverts back to control of excess air in combustion. Chimney area is calculated on the principle that a given velocity, say 15 feet per second at the prevailing temperature, is permissible. Velocities much higher than this will cause a high friction loss

in the chimney and consequently will reduce the draught. The pressure loss (P) in the chimney can be shown to be (cf. Chapter IX):—

$$= \left(1 + f \frac{4H}{D}\right) \frac{wv^2}{2g} \text{ lb. per square foot}$$

where  $f$  is the coefficient of friction,

$H$  and  $D$  are the height and diameter of the chimney in feet.

$g$  is the gravitational constant, 32.2 feet per second per second.

$w$  is the density of the flue gas in lb. per cubic foot.

$v$  is the linear velocity of the gas in feet per second, both  $w$  and  $v$  being taken at the mean temperature prevailing in the chimney.

The first term,

$$1 \times \frac{wv^2}{2g}$$

is the loss of pressure due to the velocity of escape from the top of the chimney and the second term,

$$f \cdot \frac{4H}{D} \cdot \frac{wv^2}{2g}$$

is the loss by friction.

If  $w_0$  and  $v_0$  are the density and velocity respectively under standard conditions of 60° F. and 30 inches mercury pressure, and the mean chimney temperature is  $t^\circ$  F., the expression becomes

$$P = \left(1 + f \frac{4H}{D}\right) 0.0000574 w_0 v_0^2 (460 + t) \text{ inches w.g.}$$

From this expression it will be seen that the total pressure loss in the chimney increases as the *square* of the velocity of the gases. The quantity of excess air used in combustion will thus have an important bearing on the available draught, in addition to its adverse effect on the chimney temperature.

As an example, there will be considered a chimney 100 feet high, 6 feet diameter designed to pass 254 cubic feet of gas per second (measured at 60° F.) with a mean temperature of 400° F. This corresponds to a value of  $v$  of 15 feet per second or  $v_0$  of 9 feet per second. It is further assumed that with these conditions, coal is being burnt with 50 per cent. excess air. Table 32 indicates the effect of excess air, if the gas temperature is assumed to be unaffected by the additional air. Excess air overloads the chimney in the same way as putting too many boilers on to the chimney.

TABLE 32

Percentage excess air	Percentage CO <sub>2</sub> in flue gas	Total gas flow cu. ft./sec.	$v_0$ ft./sec.	Loss of pressure in chimney, inches w.g.
50	12.5	254	9	0.04
100	9.5	350	12.4	0.07
200	6.4	520	18.5	0.16
300	4.7	700	25	0.30

If the excess air is due to inleakage of cold air the temperature of the gases fall. Thus, if the 254 cubic feet per second properly produced from the boiler or furnace and at a temperature of 400° F. were diluted with 266 cubic feet of cold air at 60° F., bringing the CO<sub>2</sub> content down to 6.4 per cent., the temperature of the gases would fall to 225° F. The loss of chimney pressure

would then be 0.13 inches w.g. instead of 0.16 inch as calculated in Table 32; but the draught as shown from Fig. 21 would fall from 0.55 inch at 400° F. to 0.33 inch at 225° F. The net result is a diminution of available draught of  $0.55 - 0.33 + 0.13 = 0.04$  or 0.31 inch w.g. Cold air infiltration is thus even more detrimental to draught than is the admission of excess air through the furnace.

This calculation indicates, of course, that a boiler plant which is just able to burn a given quantity of coal under good combustion conditions can no longer burn that quantity under bad combustion conditions with high excess air. The output of the plant is therefore reduced.

Bad combustion conditions may not reduce output in this way with mechanical draught, but the power required to drive the fan is in direct proportion to the volume of gases to be handled. Thus if the steam required for a properly operated installation be taken as  $1\frac{1}{2}$  per cent. of the steam generated with 50 per cent. excess air, if the excess air rises to 200 per cent., the steam used is doubled.

Having determined that the maximum of draught is available at the chimney, the next step is to make sure that there is no obstruction in the flues or plant causing undue resistance. A rough rule is to allow 0.2 inch w.g. for each 100 feet of straight flue, and 0.1 inch for each bend, but these figures will depend upon the quantity of gas flowing, and other factors (see Chapter IX). The loss of pressure due to an economiser is in general not higher than 0.25 inch w.g., and this should be checked from time to time by pressure measurements at various points in the system to ensure that accumulations of soot and flue dust are not causing undue resistance to the flow of gas. Occasionally water has been known to seep into the flues and to cause blockages.

The importance of preventing inleakage of air through faulty brickwork has been emphasised. In addition to its adverse effect on the draught available, to which reference has just been made, inleakage uses up the draught by drawing in air at places where it cannot be used for combustion, and thus starves the furnace of the combustion air there needed. It must not be forgotten that in addition to cracks, often too small to be apparent, brickwork is permeable to gases. All cracks should be carefully pointed, and the outside of the walls of boiler settings, flues and furnaces, should be coated with tar paint.

One method of searching for air leaks is to run a duck-lamp over all doubtful places and note where the flame is drawn in. Another method, applicable to certain fuels only, is to close the damper and note where smoke issues. When this method is used with coal, fresh bituminous coal would be put on to the fire immediately before lowering the damper.

The draught required depends on the nature of the fuel, on the depth of fire maintained, on the rate of combustion desired, on the design of the boiler and flues and on the resistance of ancillary plant, such as economisers, air-heaters, regenerators or recuperators, waste heat boilers and so forth.

In view of these possible divergencies in practice it is difficult to give recommended figures. The following may be taken as fairly representing average practice for an internal-flued boiler, e.g. a Lancashire boiler:—

Chimney height feet	Draught Chimney base	Draught Bridge	Combustion rate coal/sq. ft./hr.
100–120	0.65" w.g.	0.3" w.g.	20 lb.
120–150	0.9" "	0.4" "	25 "
150–200	1.1" "	0.5" "	30 "

Some coals require more draught than others. Coke, anthracite and low-volatile coals require a strong draught. Much depends upon the resistance of the firebed. With low-volatile solid fuels nearly all the air for combustion must be drawn through the firebed. With higher-volatile fuels, a considerable proportion of the combustible material is distilled off and is burnt with secondary air. Since secondary air is drawn in through openings which offer virtually no resistance to flow, the draught required to provide secondary air is a good deal less than that needed for primary air.

Low rank coals require a high draught, since the volume of products of combustion per unit of heat generated is generally greater than for the higher rank coals.

The resistance of the fuel bed has much to do with the effective utilisation of draught. The ideal is that air shall be uniformly drawn through the fire, at the same rate over each square foot of the grate, though this may be modified in chain-grate stokers where less air is needed at the extreme end of the travel.

Any segregation of the fuel on the bed will cause more air to pass through where the coal is larger, and will thus result in uneven combustion and possibly in unconsumed air passing through some portions of the grate owing to its high velocity. In effect, this will also reduce the output from the plant, since those parts of the grate where the resistance is high are not being fully utilised.

Uneven thickness of the fire will have the same effect as segregation, and if the unevenness is so great as to cause holes and bare spots, much excess air will flow through these places, there being little or no resistance to flow.

Special care should be taken when the combustion space is under suction that the fire-doors are not opened more than is necessary and that the secondary air ports or dampers are correctly adjusted. The lack of resistance to the flow of secondary air has already been mentioned; it is probable that more inefficiency occurs through lack of correct proportioning of primary and secondary air than through any other single factor in boiler plants.

Accumulations of ash, and particularly the formation of large masses of clinker, adversely affect the distribution of air, and increase the resistance of the firebed. Caking of coals on the grate also has a similar effect. Large masses of coke formed on combustion should be broken up.

A high fire-bridge will cause unnecessary resistance. The fire-bridge should be no higher than is required to keep the coal on the grate and should be streamlined, back and front.

The salient features of effective utilisation of draught may be summarised thus:—

- (1) The available draught is seriously diminished by the introduction of unnecessary excess air. If this excess air leaks in through the brickwork and is cold, the adverse effect *on the draught* is even worse than when it has passed through the furnace and is hot.
- (2) Resistances to the flow of gas in the furnace and boiler setting should be watched carefully to ensure that they do not rise beyond reasonable limits.
- (3) The objective of draught is to bring the combustion air to the right place, at the right time, and in the proper quantity. The resistance of the fuel bed has a vital bearing on this. Precautions should be taken to avoid the introduction of air at places where it cannot take part in combustion, and to maintain the resistance of the fuel bed uniform over the whole grate area.
- (4) The earlier in the system that excess air leaks in (excluding excess air which passes through the fire), the worse is its effect *on heat transmission*; the cooler gases transmit heat much less readily than the hotter gases obtained with less excess air.

# PROPERTIES AND PRINCIPLES OF UTILISATION STEAM

Pressure—Formation of steam—Work done by steam—Selection of boiler pressures—Superheated steam for power production and for process work—Use of steam in process work—Pipe lines—Return of condensate.

**T**HE greater proportion of the coal mined in this country is used for the purpose of heating water. For purposes such as central heating and many process operations the hot water is used directly. For many other purposes the water is converted into steam, which is used either for the production of power in prime movers or for the transference of heat for process work. The steam used in process work may either be indirect steam which does not come into contact with the processed liquid, or it may be direct steam blown through the liquid.

## PRESSURE

A distinction must be made between absolute and gauge pressure. A pressure gauge is a means of comparing the pressure in a vessel or pipe with the pressure of the surrounding atmosphere. A gauge freely exposed on all sides to the atmosphere reads zero. If the gauge is attached to a vessel and reads, say, 30 lb. per square inch, the pressure recorded as existing in the vessel, i.e. the "gauge pressure," is 30 lb. per square inch above that of the atmosphere.

The atmospheric pressure is measured by a barometer which gives the height of a column of mercury producing the same pressure as the atmosphere. Since the density of mercury at 0° C. is 0.49 lb. per cubic inch the barometric height in inches multiplied by 0.49 gives the corresponding pressure in lb. per square inch.

The barometric height for "normal" atmospheric pressure is 760 mm. (29.92 inches) of mercury which is equivalent to 14.7 lb. per square inch. Unless the barometer reading differs widely from "normal," or a high degree of accuracy in pressure is required, the absolute pressure can be obtained by adding 14.7 to the gauge pressure in lb. per square inch.

Zero on the scale of absolute pressure is a perfect vacuum. Normal atmospheric pressure is 14.7 lb. per square inch abs. Thus in lb. per square inch :—

Abs. pressure	0	10	14.7	100	114.7	214.7
Gauge pressure	29.92"	9.5"	0	85.3	100	200
	vac.	vac.				

## STEAM FORMATION

Steam is produced by supplying heat to water and raising its temperature until it reaches the boiling point at which temperature the water gradually changes from the liquid state to vapour in the process known as boiling. The temperature at which boiling takes place depends upon the pressure under which the steam is made. If this pressure be "normal" atmospheric, 14.7 lb. per square inch absolute, or 29.92 inches mercury barometer, the temperature is 212° F.; if the pressure is above normal atmospheric the temperature will be above 212° F., and if the pressure is lower than normal atmospheric, as at high altitudes or under a vacuum, the water will boil at a temperature below 212° F.

The water is fed to the boiler at a temperature determined by local circumstances, but it is clear that in general the feed temperature cannot be less than 32° F. at which temperature water solidifies (at atmospheric pressure) and

therefore by general consent all heat quantities contained in the steam are based on water at 32° F. which is taken as the starting point, i.e. the heat content of water at 32° F. is arbitrarily regarded as zero.

The heat required to make steam obviously depends on the quantity of steam to be made and the unit quantity in this country is the pound. If the British Thermal Unit is chosen for the unit of heat, it follows that all heat quantities in the steam will be expressed as B.Th.U. per lb.

The formation of steam consists of three operations :—

- (1) Forcing the water into the boiler against the boiler pressure.
- (2) Warming the water until it reaches the temperature at which it boils.
- (3) Evaporation, by which the water is changed into steam.

These three operations are essential for the production of steam of any quality, but often there is a fourth operation in which the steam is withdrawn from the boiler and heated further in order to superheat it.

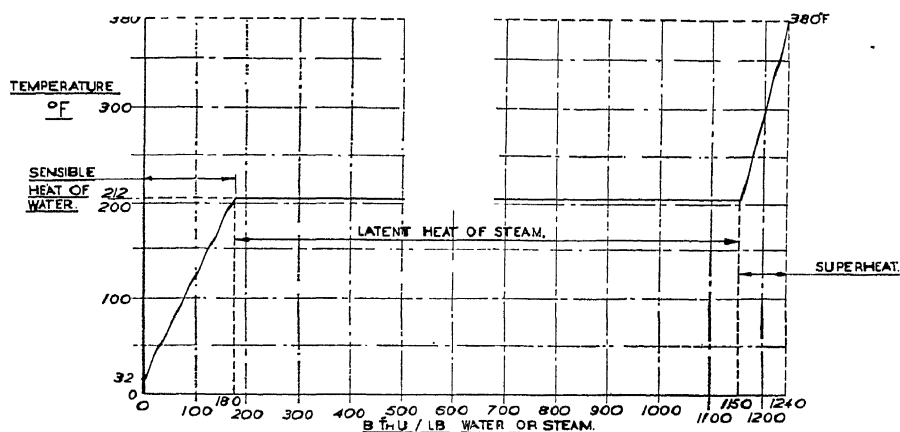


FIG. 32. Heat quantities involved in the formation of steam at atmospheric pressure.

The first operation is performed by the feed pump or injector. The energy required for this is small (the volume of 1 lb. of water being only about 0.016 cubic foot), and although in the aggregate it may be considerable, it is generally less than 1 B.Th.U. per lb.

The second and third operations occur within the boiler proper. Starting from the agreed datum temperature of 32° F., the steady application of heat produces a steady rise in the temperature of the water. If the water is mixed so that the heat supplied is uniformly distributed throughout the pound of water and the temperature is uniform, the absorption of 1 B.Th.U. raises the temperature by 1° F., the specific heat of water being unity for moderate ranges of temperature.

The temperature continues to rise with the absorption of heat until the water reaches the temperature at which it boils; this temperature, as has been mentioned above, depends upon the pressure in the boiler. If the pressure be atmospheric this temperature is 212° F. The 180 B.Th.U. which have been supplied will have produced the 180° rise in temperature from 32° to 212° F. (see Fig. 32). This heat is apparent to our senses—to our sense of warmth if we put our hand in the water, to our sense of sight if we use a thermometer. It is therefore called "SENSIBLE HEAT."

The energy supplied by the pump added to the sensible heat gives the total



amount of energy which has so far been given to the water. This quantity is called the "TOTAL HEAT OF WATER." Values of the total heat of water for different pressures and also the temperatures of boiling are given in Table 33.

When the water has reached the boiling temperature a further absorption of heat does not cause a further rise in temperature, but the water is gradually changed into steam (Fig. 32). This change in physical state from liquid to vapour may require the absorption of many heat units, but it takes place at constant temperature and therefore all this heat is hidden in the steam without producing any rise in temperature. Heat thus utilised is called "LATENT."

The heat thus rendered latent at normal atmospheric pressure is 970 B.Th.U. per lb., but the latent heat varies considerably with the pressure as shown in Table 33.

The vapour may contain drops of liquid in suspension in which case the steam is called "wet." When the last particle of liquid in the steam is converted into vapour the steam is called "dry." Any further addition of heat to the steam when dry will raise the temperature above the boiling point and the steam becomes "superheated" (Fig. 32).

Steam which has absorbed all the heat which can be absorbed without superheating is said to be "saturated" and steam which contains no particles of moisture and which will show a rise of temperature if any further heat be added is called "dry saturated."

The LATENT HEAT OF STEAM is defined as the heat necessary to convert 1 lb. of water at boiling temperature into dry saturated steam, or the heat that must be taken from 1 lb. of dry saturated steam to condense it all into water without any reduction of temperature.

All the energy which has been supplied in making 1 lb. of dry saturated steam from water at 32° F. is called the TOTAL HEAT OF DRY SATURATED STEAM and is the sum of the total heat of water and the latent heat.

Values of latent heat and total heat depend on the pressure as can be seen from Table 33 where the values are given in round figures for illustrative purposes. The maximum total heat content is at 430 lb. gauge pressure. More detailed values for the properties of steam will be found in Appendix 2.

TABLE 33. PROPERTIES OF DRY SATURATED STEAM

Pressure ins. vacuum and lb./ sq. in. gauge	Temperature of boiling ° F.	Heat in water B.Th.U./lb.	Latent heat of steam B.Th.U./lb.	Total heat B.Th.U./lb.
29.0" .. ..	77	45	1,050	1,095
27.0" .. ..	114	82	1,029	1,111
25.0" .. ..	133	101	1,018	1,119
20.0" .. ..	161	129	1,002	1,131
15.0" .. ..	179	147	991	1,138
10.0" .. ..	192	160	983	1,143
5.0" .. ..	203	171	976	1,147
Atmos. .. ..	212	180	970	1,150
100 lb. sq. in. ..	338	309	880	1,190
150 .. ..	366	338	857	1,196
200 .. ..	388	362	837	1,199
250 .. ..	406	382	820	1,202
430 .. ..	455	436	769	1,205
500 .. ..	470	453	751	1,204
1,000 .. ..	546	545	646	1,191
1,500 .. ..	597	614	553	1,167
2,000 .. ..	636	673	461	1,134
3,191.5 .. ..	705.4	902.7	0	902.7

The formation of steam under various pressures may be represented diagrammatically as in Fig. 33, in which heat in B.Th.U. is plotted against temperature in °F.

Suppose it is desired to make a pound of steam at atmospheric pressure (14.7 lb. per square inch absolute). Starting from the origin at 32° F. and zero heat supplied, the energy required to force 1 lb. of water into the boiler is too small to show in the diagram and may be neglected. The supply of 180 B.Th.U. raises the temperature of the water to the boiling point (212° F.) as represented by the line Oa. The change in physical state from water at boiling point to dry saturated steam requires 970 B.Th.U. This operation takes place at 212° F. and is depicted by a line aA which represents 970 B.Th.U. in length. The point A is therefore at 212° F. and 1,150 B.Th.U. The point a gives the total heat of

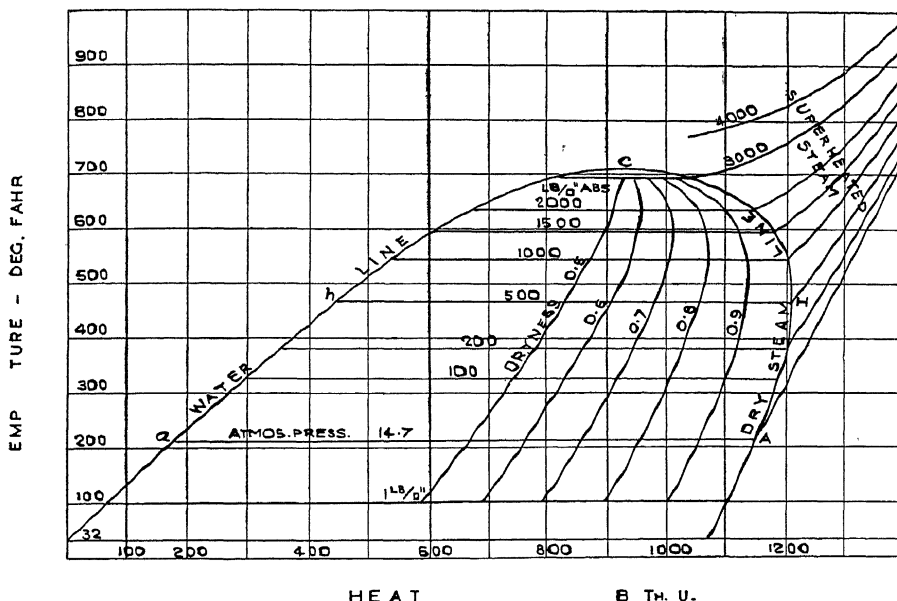


FIG. 33. Pressure, temperature, heat per pound and dryness of steam.

water, the point A the total heat of dry saturated steam and the length aA to scale gives the latent heat, all three for atmospheric pressure.

Next suppose it is desired to make steam at 500 lb. per square inch absolute pressure for which the boiling temperature is 467° F. The energy necessary to push the pound of water into the boiler is less than  $1\frac{1}{2}$  B.Th.U. and can hardly be represented on the diagram. The warming operation therefore starts from O and passes up through the point a to the point h, which has a value of 450 B.Th.U. at 467° F. The latent heat at this temperature is 755 B.Th.U., giving the line hH and the point H which is at 1,204 B.Th.U. If this process be performed for the whole range of pressures given in Table 33 the complete curve OahCHA is obtained.

Reference has been made to wet steam which contains particles of liquid in suspension. The amount of moisture is found from the DRYNESS FRACTION of the steam which is defined as the fraction of the steam which is "dry." Thus 1 lb. of steam having a dryness fraction q consists of q lb. of dry steam

and  $1-q$  lb. of liquid in suspension. Steam at 500 lb. per square inch and dryness 0.5 is represented by a point midway between  $h$  and  $H$ . Points at the same dryness can be joined together giving dryness lines as in the diagram.

**SUPERHEATED STEAM** is steam which has absorbed more heat than is necessary to make it dry saturated and which therefore has a temperature above the boiling point. The specific heat of steam depends on pressure and temperature, rising with increase of pressure and falling with increase of temperature. At pressures near atmospheric the specific heat is practically constant at 0.48 but at 500 lb. per square inch absolute the variation is from 0.74 at 480° F. to 0.51 at 1,000° F. Lines representing superheated steam are shown on Fig. 33.

It will be observed from the diagram that the water line  $OahC$  and the dry steam line  $AHC$  run together at the point  $C$ . At this point the latent heat of steam vanishes and the temperature of the point  $C$  is called the "Critical Temperature." Steam at a temperature above the critical cannot be condensed no matter to what pressure it is subjected. The pressure corresponding to  $C$  is called the "critical pressure."

## WORK DONE BY STEAM

The work which steam is capable of doing depends in the first place on the energy which is stored in it. This energy can be divided into two parts :—

- (1) The pressure energy, which the steam possesses by virtue of its pressure and volume.
- (2) The internal energy which is the difference between the total heat of the steam and the pressure energy.

### (1) Pressure Energy

The pressure energy in ft.-lb. is the product of pressure in lb. per square foot and volume in cubic feet. In B.Th.U. it is  $144 p.v/778$  where  $p$  is the pressure in lb. per square inch,  $v$  is the volume in cubic feet and 778 the mechanical equivalent of a B.Th.U. in ft.-lb. Such energy is immediately available for doing work.

The volume of 1 lb. of dry saturated steam diminishes with rising pressure, but the pressure energy does not vary very greatly. This energy can be used directly without expanding the steam and is the energy used in a direct-acting pump :—

Abs. pressure, lb./sq. in.	..	1	10	50	100	500	1,000	2,000	3,000
Vol., cu. ft./lb.	..	334	38.4	8.52	4.43	0.93	0.458	0.193	0.083
$144 p.v/778$ —B.Th.U.	..	62	71	79	82	86	85	71	46

### (2) Expansion Energy

By far the greater part of the energy is stored as internal energy. Much of this energy can be utilised by allowing the pressure of the steam to drop and the total amount of work which can be obtained theoretically by dropping the pressure of the steam is called **HEAT DROP**. It is beyond the scope of this publication to explain how such heat drops can be determined, but the following figures show how the heat drop depends on pressure drop. If dry saturated steam at 500 lb. per square inch absolute is expanded in a perfect engine to  $p$  lb. per square inch absolute the heat drops are as follows :—

	500	100	50	14.7	10	3	1	
Heat drop.	0	129	175	253	272	339	391	B.Th.U. per pound

If dry saturated steam at pressure  $p$  lb. per square inch absolute is expanded to 1 lb. per square inch absolute the heat drops in B.Th.U. per lb. (cf. Appendix 3) are :—

	3,000	1,500	1,000	500	100	50	10
Heat drop..	381	431	423	391	293	248	140

The work done by 1 lb. of steam is thus seen to depend on the pressures of supply and exhaust.

The number of B.Th.U. required per lb. of steam reckoned from water at 32° F. has already been considered and values have been given in Table 33, but if the steam is exhausted at atmospheric pressure its temperature will be 212° F., and it should be possible, theoretically at any rate, to return to the boiler the lb. of condensate at or near this temperature and thus reduce the number of B.Th.U. required per lb. of steam by 180. If the plant were a condensing plant with a vacuum of 28 inches mercury for which the temperature of condensation is 101° F. the reduction would be 101 — 32 = 69 B.Th.U. per lb.

Two important quantities in a power plant are the amount of work obtained per lb. of steam and the number of heat units required to make that lb. of steam, and the ratio of these two quantities gives the measure of the efficiency of the plant in performing the task of converting heat into work. This efficiency is called the THERMAL EFFICIENCY. If the heat equivalent of the work obtained from 1 lb. of steam is  $U$ , which in a perfect engine without heat or steam leakage is the "heat drop," and if  $H$  is the total heat of the steam as delivered from the boiler reckoned above water at 32° F. and  $t_w$  is the temperature of the hot well containing the condensate then

$$\text{Thermal efficiency} = \frac{U}{H - (t_w - 32)}$$

The thermal efficiencies corresponding to the heat drops given above when steam exhausts at 1 lb. per square inch absolute are :—

Boiler pressure—lb./sq. in.	3,000	1,500	1,000	500	100	50	10
Heat drop—B.Th.U./lb. . .	381	431	423	391	293	248	140
Heat supplied—B.Th.U./lb.	970	1,113	1,135	1,134	1,117	1,107	1,073
Efficiency—percentage . .	39.3	38.7	37.3	34.6	26.3	22.4	13.1

These figures explain in part the modern tendency to raise boiler pressures in power plants.

## WIREDRAWING

If steam is expanded through a valve or orifice it does no work and therefore loses no heat energy. Suppose dry saturated steam at 100 lb. per square inch is expanded to atmospheric pressure. Table 33 gives the total heat at 100 lb. as 1,190 B.Th.U. per lb. at a temperature of 338° F. The total heat of steam at atmospheric pressure is 1,150 B.Th.U. There will therefore be a surplus heat of 39 B.Th.U. in the expanded steam, which can only superheat the steam. At atmospheric pressure and moderate temperature, the specific heat of steam may be taken as 0.49, so that there will be 80° F. of superheat temperature. The expanded steam will thus have a temperature of 292° F. Expansion through a valve or orifice when no work is done is called "throttling" or "wiredrawing."

## UTILISATION OF STEAM

Steam is required for three primary purposes :—

- (1) For the production of power.
- (2) For use as a medium for transmitting heat for process work.
- (3) As a physical or chemical agent in industrial operations.

The same steam may be used in sequence for more than one of these operations. Thus, it may be generated under high pressure and used first for the production of power in pass-out turbines, the exhaust steam at a suitable temperature being then used for heating or for chemical purposes.

As examples of the third use of steam may be cited the use of direct steam in distillation (a physical use) and the use of steam in gas producers or in the production of hydrogen by the iron-steam process (chemical uses). Steam used for cooling fire-grates or preventing clinker formation comes under this third category.

## SELECTION OF BOILER PRESSURE

In any use of steam a decision must be taken as to the pressure at which the steam should be generated, the pressure at which it is to be used and whether it should be saturated or superheated.

(a) *For Power Production.* Some indication has been given that the modern tendency in large power generating plants is to increase the boiler pressure but, as efficiency depends upon other factors as well as boiler pressure, it is impossible to give any simple rule for the selection of boiler pressure for a particular plant. Power plants of over 50,000 kW capacity, both in this country and in America, work on pressures varying between 1,900 and 265 lb. per square inch. In almost every case there is a high degree of superheat, the temperature of the steam being between 650° and 950° F. For the higher pressure plants some degree of reheating the steam after partial expansion may be necessary, otherwise the steam becomes too wet during expansion.

The degree of superheating, reheating and feed heating all affect the efficiency of the plant and make any suggestion of optimum pressure misleading.

Higher pressures can be used in turbine plants than in reciprocating engines because turbine plants do not limit the ratio of expansion which can be usefully employed and, furthermore, are specially adapted for using large volumes of steam such as must be dealt with if the steam is expanded to low pressures which is an essential condition for high efficiency. Reciprocating engines, even if compounded in two or three cylinders, impose a limit on expansion which turbines do not. It is therefore not possible to work over so wide a pressure drop in a reciprocating engine as in a turbine.

The type of prime mover employed may modify the form of heating surface required to utilise the heat in the products of combustion, e.g. from a boiler. Thus in the modern high efficiency power plant a form of feed water regeneration is incorporated, using steam bled from the main turbine.

In such a plant the feed temperature may nearly approach the temperature of the water in the boiler and the boiler heating surface proper has then only to transmit the latent heat of steam to the heated water. Much of the heat that would have been absorbed from the hot gases by a feed water economiser can now be utilised for preheating the air supplied to the furnace.

This necessitates the provision of air-heating equipment, but since the upper limiting temperature of the air supplied to a mechanical stoker (if operating troubles are to be avoided), is about 350° F., it is easy to conceive circumstances in which the air temperature must be allowed to exceed this, or a high gas outlet temperature be permitted. Pulverised fuel burners, however, can use hot air up to temperatures of 400°–500° F., and when power plant design conditions arise as depicted above pulverised fuel firing is a rational solution of the problem.

(b) *For Process Work.* For process purposes, steam should not be generated at a higher pressure than is dictated by the temperature at which it is required to give up its heat. Additional costs are involved in operating at higher pressures, both in the design of the boiler and in increased plant to recover heat from the flue gases which of necessity must leave the boiler at a higher temperature. It is thus desirable to select the lowest boiler pressure for process work consistent with securing an adequate steam temperature.

Advantage can, however, be taken of higher steam pressures and tempera-

tures to provide power as a by-product. If the steam is generated at elevated pressures (200 lb. per square inch, for example) the excess pressure can be used to generate power, leaving the exhaust, which may still be at 30 lb. pressure, for use in process work.

### SUPERHEATED STEAM FOR POWER PURPOSES

For reciprocating engines, one reason for superheating steam is to reduce condensation in the cylinder. The temperature of the steam supplied to a reciprocating engine should therefore be as high as is permitted by the practical consideration of lubrication. Modern lubrication technique has removed in a very large measure the essential difficulty of the use of superheated steam in reciprocating engines.

When the exhaust steam from engines is led to a process for which it is essential that the steam must be clean, the steam must reach the engine in the saturated state, since any superheat, however small, demands lubrication.

In turbines it is possible to allow the steam to expand more than is practicable in a reciprocating engine. This means that the exhaust temperature will be lower and if saturated steam had been used much of the steam would have condensed. It has been found by many years of experience that 13 per cent. wetness at exhaust is the greatest amount of condensation that is permissible without undue blade erosion. Whatever pressure is selected, sufficient superheat should be added to prevent the steam leaving the last row of turbine blades more than 13 per cent. wet.

Expert advice on this subject can be obtained from the manufacturers of power station boilers and of turbines. Much useful information is also available in the Proceedings or Transactions of the Technical Institutions.

As a general statement the saving in fuel in prime movers due to the first 100° F. of superheat may be of the order of 6 or 7 per cent.; the saving due to the next 100° F. of superheat will be somewhat less, say 4 to 5 per cent. and so on.

The maximum superheat temperature needed by those industrial plants exhausting from turbine to process is about 750°–800° F., though much higher temperatures have been used in power stations.

The primary saving in fuel by the use of superheated steam is quite considerable when the steam is used for power generation. The quantity of steam for a given output of power is considerably reduced. There is, therefore, less work to be done by the feed pump, by the condenser and by the cooling water circulating pump.

Relative average steam consumptions when using saturated steam and superheats of 100° F. and 200° F. are given in Table 34.

TABLE 34

Type of engine	Steam consumption—lb. per I.H.P./hour.		
	Saturated steam	100° Superheat	200° Superheat
Simple non-condensing .. ..	29–45	20–38	18–35
Simple non-condensing Corliss ..	26–35	18–30	—
Compound non-condensing .. ..	19–28	15–25	13–22
Compound condensing " .. ..	13–22	10–20	9–17
Simple duplex steam pumps .. ..	120–200	80–160	—
Turbines non-condensing (kWh.) ..	28–60	24–54	21–48
Turbines condensing (kWh.) .. ..	12–42	10–38	9–34
Simple non-condensing locomotive. .	27	23	19·5

Reciprocating engines and turbines show a reduction in steam consumption of about 1 per cent. for every 10°–15° F. of superheat for a given power developed. This is not an equivalent gain in overall thermal efficiency (i.e. it is not an equivalent saving in fuel) because superheated steam requires more heat to be put in it initially.

For power generation substantial advantages have thus been shown to be effected by the use of superheated steam.

The superheater may also be a means of increasing boiler efficiency in various ways: firstly by reducing the stack temperature due to the abstraction of heat in the superheater and secondly by increasing the prime mover efficiency, and thereby reducing the work to be done by the boiler plant for a given output of power. This second effect will be particularly valuable if the boiler is overloaded when operated without a superheater. Thirdly, the boiler auxiliaries will have less work to do for a given output.

### THE USE OF STEAM IN PROCESS WORK

The object of steam-heated process plant is to transfer a definite quantity of heat from the steam to the material being processed. It is usually essential that the temperature of the processed material be accurately maintained within close limits.

Process heating can be divided into the following broad groups:—

- (1) Evaporation.
- (2) Raising the temperature of a processed substance :
  - (a) By means of heating surface.
  - (b) By direct contact.
- (3) Maintaining the temperature of a processed substance by compensating for the heat lost by radiation, etc., and absorbed by endothermic reactions:
  - (a) By means of heating surface.
  - (b) By direct contact.
- (4) Distillation.

#### (1) Evaporation

Evaporation for the purpose of concentrating a solution is a widely used industrial process requiring much steam. It differs from distillation in that its goal is the production of a concentrated solution, whereas the aim in distillation is the production of condensed distillate.

The steam is required to supply heat to the process. This heat consists of:—

- (1) Heat required for warming the liquid up to the temperature of evaporation, and
- (2) Heat required for evaporation.

The liquid may be warmed either by live steam or by exhaust from the evaporator, but the live steam applied to the evaporator will have to provide not only the latent heat of the liquid evaporated but heat lost by radiation, etc., from the outer surface of the plant.

From the principles of heat transmission (Chapter VIII) it will be evident that there must be some temperature difference between the (indirect) heating steam and the liquid heated, if there is to be an adequate rate of heat transfer. The temperature of the steam must, therefore, always be a few degrees (say 5°–20° F.) higher than that of the boiling liquid. If this liquid is water, it follows that the pressure of the steam must be greater than the pressure at which evaporation is taking place.

The total heat of the condensate is above the total heat of water at atmo-

spheric pressure, and therefore some of the water will change to steam when the pressure on the condensate is reduced to atmospheric.

Whether the steam used for evaporation, or for any form of indirect heating, should be saturated or superheated depends on the principles of heat transmission. When saturated steam unmixed with any permanent gas is fed into the heating jacket or tubes of a vessel, the steam comes in direct contact with the heating surface. It is immediately chilled, loses its latent heat and is condensed. The film of water runs down the surface and more steam immediately flows in to take the place of that which has been condensed. Thus the steam is condensing in contact with the water film adhering to the heating surface and heat transfer is rapid.

If the steam is mixed with permanent gas, there is set up a gaseous film in streamline motion between the film of condensed water and the main body of the steam. The steam must diffuse through this gas layer to reach the surface (a very slow process) or heat must be conducted through the gas layer; gases are bad conductors of heat. The presence of gas therefore slows down the rate of heat transfer seriously.

If the vessel is heated by a permanent gas the streamline gaseous layer is still present (see Chapters VIII and IX) and heat must be transmitted through it—a slow and inefficient process. Superheated steam behaves as a permanent gas and is thus undesirable for most operations in which steam is used for indirect heating. This is discussed in more detail in Chapter XXIII.

The vapour which leaves an evaporator carries away its latent heat and in many plants this latent heat is lost. The vapour is condensed in a separate condenser and the heat in the vapour is transferred to the cooling water or to the surrounding air if the condenser is air cooled. It is possible sometimes to conserve this heat by using a second evaporator as a condenser to the first, a third as condenser to the second, and so on until the heat has been reduced to such a low temperature that it is no longer economically useful. This is known as multiple effect evaporation.

The essential of multiple effect evaporation is that each evaporator must evaporate at a temperature lower than that of the heat supplied to it. This may be illustrated by assuming that each in a series of evaporators is evaporating water. The necessary drop in temperature through the series then demands a corresponding drop in pressure.

To illustrate the principle, let it be assumed that steam is supplied to an evaporator at 100 lb. absolute pressure; it then has a temperature of 328° F. and a latent heat (available for heating in the evaporator) of 889 B.Th.U.

If the evaporator operates at 312° F. under a pressure of 80 lb., the latent heat of the water to be evaporated is 901 B.Th.U. per lb.

Weight of steam at 100 lb. required to provide 901 B.Th.U.

$$= 901/889 = 1.013 \text{ lb.}$$

Heat contained in the vapour leaving the evaporator is 901 B.Th.U. per lb.

Thus the expenditure of  $1.013 \times 889$  B.Th.U. has resulted in driving off 1 lb. of vapour having a latent heat content of 901 B.Th.U.

This pound of vapour can be passed forward as the heating agent in a second evaporator. If this second evaporator works at 40 lb. pressure, the conditions are:—

pressure ..	..	..	..	..	40 lb.
temperature ..	..	..	..	..	267° F.
latent heat ..	..	..	..	..	934 B.Th.U.

The available heat, namely 901 B.Th.U., can then evaporate in the second evaporator, 901/934, or 0.965 lb. of water. This has been effected without any additional heat, so that the heat in the original steam has virtually been used twice over.



The 901 B.Th.U. now in the form of steam at 40 lb. pressure can then be passed to a third evaporator, and so on.

In this illustration all losses have been neglected and it has been assumed that the heat supplied in evaporation is fully returned in the subsequent condensation and can be used again without loss. Actually small losses will occur, but this principle of multiple effect evaporation, well known in chemical engineering, may be very valuable.

If the circulation of the process material is sluggish and there is a considerable depth of liquid in an evaporator, it is possible for the lower layers of material to be raised to a temperature considerably in excess of the upper layers. This is due to the hydrostatic head of the liquid exerting a pressure on the liquid in the bottom of the vessel.

It might be thought that a high temperature on the steam side of the heating surface would increase this danger, but often this is not so. With a high temperature on the steam side the higher rate of heat transfer increases the circulation of the boiling liquid due first to convection currents and second to the mechanical drag of the bubbles of vapour acting on the liquid.

This increased circulation has beneficial effects. It greatly minimises the danger of bottom overheating because with violent circulation local stagnation is unlikely. By increasing the rate of flow over the heating surface the rate of heat transfer is much increased. The fact that the liquid is full of bubbles clearly reduces its mean specific gravity. A given depth of violently boiling liquid will therefore exert a lower hydrostatic pressure than the same depth which is boiling sedately.

## (2) Raising the Temperature of a Processed Substance

(a) *By means of a Heating Surface.* The temperature of a substance can be raised by means of a heated surface with which it is in contact. The discussion relating to evaporators applies here equally with an important addition. By the use of saturated steam a limit can be set to the temperature to which the material can be raised if, by negligence, the steam is left on too long. With superheated steam the temperature of the heating surface is no higher than with saturated steam during the temperature raising process, but, whereas with saturated steam the heating will stop when the temperature has risen to the saturated steam temperature, when using superheated steam it can rise, possibly undesirably, to the temperature of the superheated steam. Superheated steam is here clearly not to be recommended.

(b) *By Direct Contact.* Many processes use steam injectors to raise the temperature of the process substance. There may sometimes be a legitimate use here for superheated steam. For a given quantity of water superheated steam will bring in more B.Th.U. and this may be important. On the other hand, some parts of the material will be in contact with and be temporarily raised to the temperature of the superheated steam, with possibly detrimental results.

Superheated steam blown into a liquid may have insufficient time to desuperheat itself and condense, so that some breaks the surface and is wasted. This is particularly likely to happen when medium pressure steam is blown into a shallow vat or tank. If dry, saturated steam at 30 lb. per square inch is blown into a vat containing 4 feet 6 inches of liquid, there will be a pressure drop in the blower of 28 lb., and the steam will be superheated by 38° F. It is very unlikely that 4 feet 6 inches of hot liquid can remove 38° of superheat and condense the steam during its passage through the liquor.

## (3) Maintaining the Temperature of a Processed Material

The principles are the same as those just stated, except that the reasons for

not using superheated steam are even stronger. Maintenance of temperature means maintenance of temperature and a plant that can produce overheating by the use of superheated steam is clearly unsatisfactory.

From the foregoing it can be stated quite categorically that superheated steam for most process purposes is not only of no benefit but may be undesirable.

There is one possible case where superheated steam might usefully be applied. This is in a piece of process plant where only the superheat is used, the steam passing through the plant into the process main. This enables one piece of high temperature plant to operate in conjunction with a much larger amount of low temperature plant on a single low pressure steam supply.

#### (4) Distillation

(a) *By indirect Steam.* This is similar in general characteristics to evaporation. The heat provided by the steam must be sufficient to heat the liquid to its boiling point (if this has not been already done outside the still), to supply the latent heat of evaporation, and to compensate for any losses of heat that may occur.

The pressure of the steam should be just high enough to provide steam at the temperature at which the liquid boils, plus an additional 5°, 10°, or 20° F., according to circumstances to provide the necessary temperature gradient for the passage of heat (cf. Chapter VIII).

(b) *By Open Steam.* Direct steam is often used for distillation, steam being passed directly through a liquid so that the more volatile constituents therein are distilled at a temperature below their normal boiling point. The steam and the distillate vapour each exert pressure, and the mixture boils when the sum of the partial pressures equals the external pressure. Generally, indirect heat is also supplied by means of jackets or coils, or by preheating the feed up to its boiling point.

Substances of high boiling point may thus be distilled at relatively low temperature; aniline, for example, boils at 180° C. (356° F.) at 760 mm., but distils in a current of steam at 98.5° C. (210° F.) at the same pressure.

The pressure may be lowered by working under a vacuum with further consequent reduction of the temperature at which distillation occurs.

Steam distillation yields a condensate consisting of water and the volatilised substance or substances in proportions dependent both on their respective vapour pressures and molecular weights.

$$\begin{aligned} & \text{i.e. at } t^{\circ} \text{ C., } \frac{\text{Weight of steam in vapour}}{\text{Weight of volatile substances in vapour}} \\ &= \frac{(\text{Vapour pressure of steam at } t^{\circ} \text{ C.}) \times \text{mol. wt. of steam}}{(\text{Vapour pressure of vol. subs. at } t^{\circ} \text{ C.}) \times \text{mol. wt. of vol. subs.}} \end{aligned}$$

As general examples may be cited the stripping of crude benzole from wash oil in the carbonisation industry, the deodorisation of fatty oils for the food industry, the steam distillation of fatty acids, essential oils, aniline and other organic substances.

Steam distillation is particularly useful in processing materials subject to decomposition at temperatures approaching their normal boiling point. Excluding a special case dealt with in Chapter XXIII the substance to be steam distilled must be immiscible with water.

The steam performs a dual function.

- I Its presence lowers the partial pressures of the substances distilled, thus reducing their effective boiling point and so facilitating distillation.

- II The sensible heat in the steam above the effective boiling point of the mixture serves to supply some of the latent heat required to volatilise the constituents to be distilled. Normally a considerably greater amount of latent heat is required than is available from the open steam and, as indicated above, this is supplied indirectly.

Steam distillation operations may be divided into two classes :—

- (1) Where the boiling mixture in the still contains liquid water.
- (2) Where the steam is not allowed to condense in the still.

There is obviously no economic advantage in the use of superheated as opposed to saturated steam when liquid water exists in the still. There are, however, some instances when condensation of water is not permissible (class 2) where superheated steam may be advantageously employed as the superheat permits a higher temperature to be attained at the point of primary vaporisation.

Examples indicating the conditions and the lines upon which the economics of the use of direct steam for distillation may be considered are given under "Distillation" in Chapter XXIII.

### PIPE LINES

Steam is conducted from the boiler to the place where it is used in pipe lines. Heat is lost from these pipes and in consequence superheated steam will lose part or all of its superheat, being ultimately converted into saturated steam, and saturated steam will become wet.

The radiation loss from steam pipe lines is mainly determined by the degree of insulation. The better the insulation the greater the economy due to reduction of heat loss. It is very important to insulate the flanges as well as the pipe itself.

If the steam be sufficiently superheated, it will reach the process plant dry. If initially it is dry and saturated, or wet, it will reach the process plant wet. For a given pipe the loss of heat is likely to be somewhat less with superheated steam than with saturated steam. There is thus some saving in superheating through reduced loss of heat in the transmission pipes, and there is a considerable saving in traps and their ancillary piping.

Two other advantages possessed by superheated steam are :—

- (a) There is less wear and tear on valves and fittings generally.
- (b) Speeds about 50 per cent. higher can be used in pipe lines with superheated steam than with saturated steam. In new plants this would mean smaller pipe lines ; in existing plants, the required quantity of heat can be conveyed through the pipe lines for a smaller drop in pressure.

### CONDENSATE

The exhaust from prime movers or from process operations will consist of a mixture of steam and hot water at a temperature corresponding to the back pressure against which the exhaust is working.

If exhaust steam from prime movers is led directly into a water supply tank in order to heat the water, a back pressure is created which reduces the efficiency of the engine or turbine. Consequently, a condenser is frequently used for the purpose of condensing the steam and of reducing the back pressure. The quantity of heat abstracted in the condenser will depend upon the dryness fraction of the exhaust steam as well as upon its pressure.

If, for example, the exhaust from an engine is 70 per cent. dry at an exhaust pressure of 4 lb. per square inch absolute, the heat rejected will be in three forms \* :—

\* At 4 lb. abs. the boiling temperature of water is 153° F., the sensible heat in the water is 121 B.Th.U./lb. and the latent heat is 1,006 B.Th.U./lb.

Sensible heat in water at 153° F.—30 per cent. of 121 = 36 B.Th.U.

Latent heat in 70 per cent. steam—70 per cent. of 1,006 = 704 B.Th.U.

Sensible heat in water from the

steam—70 per cent. of 121 = 85 B.Th.U.

Total = 825 B.Th.U. per lb.  
of exhaust

If the steam was generated without superheat at 100 lb. gauge pressure, it will originally have contained 1,190 B.Th.U. per lb. Nearly two-thirds of the heat in the steam would therefore be contained in the exhaust.

Similar considerations apply to the exhaust or hot water from process work. It is therefore of great importance that the heat contained in this exhaust shall be utilised.

When steam condenses in a pipe line or process vessel, the hot water must be removed without loss of steam. This is usually done by means of an automatic device called a steam trap.

The condensate and the heat it contains are valuable. It should, therefore, be returned to the boiler whenever possible. When it cannot be returned to the boiler, it should be used as hot process or washing water. If by any chance the condensate is corrosive, or if it contains oil, its use may be impossible, but even then the heat it contains can often be used by passing it through a heat exchanger.

## FLASH

The hot water leaving steam traps may be at temperatures above 212° F., and at corresponding pressures above atmospheric. Thus, if steam is being used at a pressure of 100 lb. per square inch gauge, the water condensed from it at this pressure will be at 338° F. and will have a total heat above 32° F. of 309 B.Th.U. per lb. If it is to be run back into the boiler feed tank at atmospheric pressure its temperature cannot be above the boiling point of water at that pressure, namely 212° F. Heat must be abstracted from the water to cool it from 338° F. to 212° F. If the trap exhausts to atmosphere, i.e. discharges the water through a pipe direct to the boiler feed tank, part of the water will flash into steam and be lost. The heat thereby lost is as follows :—

Heat in 1 lb. of water at 338° F. .. 309 B.Th.U.

" " " 212° F. .. 180 " "

Heat loss per lb. of exhaust 129

TABLE 35. FLASH STEAM

Condensate		Percentage condensate flashed off when pressure is reduced to :—					
Pressure lb/sq. in. gauge	Temp. ° F.	40 lb/ sq. in.	20 lb/ sq. in.	10 lb/ sq. in.	Atmos.	10" vac.	20" vac.
200	388	11.5	14.3	16.2	18.8	20.5	23.2
150	366	9.0	11.8	13.0	16.4	18.2	20.9
100	338	5.8	8.6	10.6	13.3	15.1	17.9
80	324	4.2	7.1	9.1	11.9	13.7	16.5
60	308	2.3	5.2	7.3	10.0	11.8	14.7
40	287	—	3.0	5.0	7.8	9.7	12.6
20	259	—	—	2.1	5.0	6.8	9.8
10	240	—	—	—	2.9	4.8	7.8
0	212	—	—	—	—	1.9	5.0

As the latent heat at 212° F. is 970 B.Th.U., a quantity of water will be evaporated as "flash" steam, amounting to  $129/970 = 0.133$  lb. per lb. of condensate. Some further data on this subject are contained in Table 35.

A loss from the evolution of flash steam arises wherever a steam trap is discharging above atmospheric pressure. The extent of the loss depends on the pressure and if, for example, the trap were working at 25 lb. gauge pressure, the loss would be  $236 - 180 = 56$  B.Th.U. per lb. of steam or water.

The loss can be avoided in various ways :—

- (1) The water leaving the traps can be passed through a heat exchanger in counter-current with the make-up feed water.
- (2) If low pressure steam can be utilised the water can be discharged to receivers connected to the low pressure steam supply, the flash steam being then utilised for the supply.
- (3) The water may be collected in a tank under pressure in which the make-up feed is introduced to condense all flash steam.

The generation of flash steam by the reduction of pressure on hot water is used in steam accumulators (Chapter XXIII).

Another application of the release of heat by pressure reduction is in the cooling of liquids. If an aqueous liquor at, say, 200° F. is to be cooled to 150° F., this can be readily accomplished by spraying the hot liquor into an empty vessel attached to a condenser and vacuum pump—the vacuum being such as to correspond to a liquor boiling temperature of 150° F. At the same time as cooling there will be a small concentration which may save evaporation at a later stage in the process.

The practical utilisation of steam is discussed at greater length in Chapter XXIII.

## CHAPTER VIII

## HEAT TRANSMISSION

Methods of heat transfer—Transmission of heat through a boiler plate—Conduction—Overall conduction coefficients—Conduction through composite walls and pipe coverings—Convection—Natural convection—Forced convection—Convection effects in banks of tubes and in single tubes—Heat transfer from superheated steam—Convection in liquids—Heat transfer from condensing vapours—Boiler circulation—Radiation—The laws of radiation—Radiation from a coke fire—Gas radiation, including methods of calculation—Radiation from luminous flames—Heat exchangers, regenerators and recuperators—Heat losses from external surfaces—Heat loss through a furnace wall—Practical aspects of heat transmission in boilers.

**T**HE process of heating involves three stages. Firstly, the heat must be generated by efficient combustion as described in Chapters IV–VI. Secondly, the heat so generated must be transmitted as efficiently as possible to the heat-absorbing surfaces. Thirdly, the heat must be conserved in the heating appliance to do useful work, and its transmission to those parts of the appliance from which it may be dissipated must be reduced to a minimum. In addition to the loss of heat so occurring through the structure of the appliance and its foundations there are the combustion losses which have already been discussed.

## MODES OF HEAT TRANSFER

Heat may be transferred in three different ways, by conduction, convection and radiation. All three methods depend upon a difference of temperature in the heat transmitting media, and heat is always transferred from the warmer to the cooler body.

*Conduction.* The simplest form of heat conduction takes place in a solid, say a boiler plate or a furnace wall. One surface may be hot and the other cooler. The quantity of heat conducted through the body in a given time is approximately directly increased by an increase in the difference of temperature between the hot and cold surfaces. It is further proportionately reduced by an increase in the thickness of the plate or wall through which the heat is being conducted. It is proportionately increased with the dimensions of the surface normal to the direction of flow, and finally the rate of heat conduction depends upon the nature of the material involved. The familiar reference to good conductors such as metals, and bad conductors such as firebrick, lagging compounds and boiler scale can be reduced to a form of expression suitable for use in heating problems, by stating the number of heat units (British Thermal Units) which can be conducted in one hour through a slab of the material 1 square foot in section and 1 foot thick when there is a difference of temperature of 1° F. between the hot and the cold face. The conduction must be effected when temperatures are steady, referred to as “in the steady state”, and under these conditions the number of heat units conducted hourly is referred to as the “thermal conductivity” of the material. It is necessary to distinguish between the steady or uniform flow of heat and fluctuating flow, since in fluctuating flow heat may be given to, or taken from, the body of the slab. This arises from the fact that the material has a capacity for absorbing and storing heat, and the temperature gradient in the slab is accordingly disturbed by this capacity. In heating it absorbs heat; in cooling heat is given up.

Heat transfer by conduction can take place in still liquids and gases, but this mode of heat transmission, apart from the stagnant film of gas collecting on solid surfaces, is of secondary importance in industrial practice.

*Convection.* In liquids and gases the moving particles transmit heat to one another by contact if they differ in temperature, and they transmit heat by

contact when they are brought against the surface of a solid. This transmission of heat by the motion of the fluid against a solid is referred to as convection transfer, since it is brought about by the flow or convection of the fluid. Where the motion of the fluid results from its natural buoyancy arising from heating, expansion and consequent fall of density, the phenomenon is described as free or natural convection. When the motion is maintained mechanically, as for example by a fan, it is referred to as forced convection.

*Radiation.* Conduction and convection represent the transfer of energy from particle to particle by direct contact. Energy is also emitted from a hot body and can be transmitted in the form of radiation to another separated from it by a suitable medium. The heat emitted from an open firebox door is a case in point. A hot solid body sends out radiation over a wide range of wave lengths from the smallest to the largest, but the intensity of the radiation differs greatly for the various wave lengths to which the human eye is sensitive. The wave length of violet light is about 4,000 Angstrom units, Å, and red light ends in the region of 7,000 Å, an Angstrom unit being  $10^{-10}$  metres. Longer waves than 7,500 Å are known as the infra-red, and within 7,500 to 53,000 Å, falls the range of thermal radiation which is of practical importance in industrial heating. Radiation is also emitted by hot furnace gases.

If the phenomena of conduction and convection on the one hand are contrasted with thermal radiation on the other, it is found that the former are affected by temperature difference and very little by temperature level, whereas radiation increases rapidly with increase in temperature level. Accordingly at very low temperatures conduction and convection are the major contributors to the total heat transfer; at very high temperatures radiation is the controlling factor.

Heat transmission by conduction occurs in the passage of heat through a boiler plate and its adherent resistant layer of soot and scale; into the mass of the charge in a furnace; and through the hearth, walls and crown of a furnace chamber or firebox.

Convection transfer is bound up ultimately with the mechanism of flow of fluids, which is discussed in detail in Chapter IX, and is an important factor in the heating of waste heat fire-tube boilers, and in any form of heat exchanger in which a fluid flows over the heat-exchanging surface, for example in superheaters, economisers and air heaters. The velocity of flow of the gases is of much significance in these conditions.

Radiation is universal in its occurrence, being operative in addition to natural convection from the relatively cool surface of a lagged steam pipe, as well as from the incandescent surface of a fuel bed or a refractory wall.

## TRANSMISSION OF HEAT THROUGH A BOILER PLATE

It is now possible to construct a picture of what happens when a stream of hot combustion gases heats a boiler tube (Fig. 34). The boiler shell or tube does not consist merely of a plate of mild steel, but the surface adjacent to the hot gases is covered with a layer of soot and oxide of iron, and on the evaporation side is covered with a thin layer of boiler scale, which may consist mainly of mineral deposits from the boiler water.

The heat of the combustion gases is transmitted to the surface of the thin layer of soot and deposit by radiation and convection, since radiation is emitted by the hot gases, and the scrubbing action of the flowing gases further increases the total heat transfer. At the surface of the boiler tube the gases directly in contact with the layer of soot and scale are virtually stagnant by reason of the resistance to the flow of gas of the surface presented. There is thus formed a stagnant film of gas through which the heat must be transmitted by conduction. Since gases have a low thermal conductivity, this film offers a consider-

able resistance to the flow of heat, out of all proportion to its thickness (see Chapter IX).

Heat is again transmitted by conduction through the badly conducting fouling layer on the hot surface of the plate, then through the plate which is a good conductor and through the second fouling layer and finally to the water, where ebullition and evaporation of steam take place.

In boiling a liquid at a given pressure, as the temperature difference between heating surface and liquid is increased, the heat flux at first increases slowly, then more rapidly, goes through a maximum and subsequently decreases as the heat transfer surface becomes insulated with a film of vapour. The character of the heat transmission on the water side must accordingly be influenced by the convection conditions, and will thus depend on the character of the water

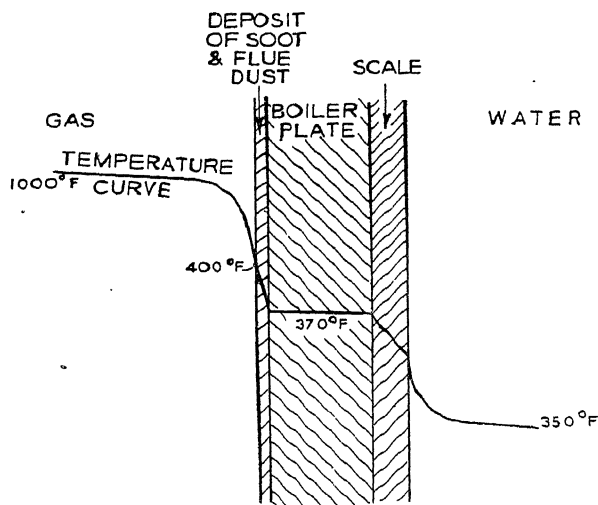


FIG. 34. Diagram illustrating heat resistances and their effects on temperature at a boiler heating surface. Scale may cause the temperature of the boiler plate to rise much above the figures stated on the diagram, particularly where the plate is subjected to direct radiation from the furnace.

circulation. Again a "film" factor on the water side of the plate becomes operative.

It will be appreciated in comparing the performance of various types of boilers and in investigating conditions in any one boiler that the individual layers of media more or less resistant to the heat flux cannot be considered independently, for the reason that the temperature differences which govern the heat transmission in the various layers cannot be known with any accuracy. Accordingly it has become the practice to use the concept of a "surface conductance" or "film" coefficient, which may be applied to include the stagnant layer of fluid and the fouling condition. The "surface conductance" then becomes a coefficient defining the rate of heat flux for each degree of temperature difference between the hot gases and the boiler plate on the one side and the boiler plate and the water in the boiler on the other. These surface conductances are combined with the conductances of the various solid layers so that an "overall surface conductance" or coefficient can be arrived at to define the practical character of the plate in respect of heat transmission.

An allied conception is the "fouling factor" or "dirt factor" which is



equivalent to a decrease in the "film coefficient" or surface conductance to correspond to the increased resistance to heat flow. The use of this factor in computing heating surface has proved extremely valuable in practical work.

It is inevitable that the conservation of fuel should be bound up with the measurement of heat quantities. Accordingly in order to take the study of the subject of heat transmission beyond the elementary descriptive stage so far attempted, recourse to the methods of computation are necessary. The subject may be complicated and mathematical, but such a treatment is necessary, since no alternative is available.

## CONDUCTION

### APPLICATION OF THE LAW OF CONDUCTION TO TRANSMISSION OF HEAT THROUGH WALLS

The law of conduction, which has been described in words earlier in this chapter, can be expressed for the simplest case of heat flowing through a parallel plate (Fig. 35) at right angles to the sides of the plate, as follows:—

$$q \text{ (in heat units per hour) varies as } \frac{(t_1 - t_2)A}{L}$$

$$\text{or } q = \frac{k(t_1 - t_2)A}{L} \quad (1)$$

where,  $k$ , the constant of proportionality, varies for different materials and is known as the thermal conductivity of the material.

For the conduction of heat per unit area of cross section

$$\frac{q}{A} = H = \frac{k(t_1 - t_2)}{L} \quad \dots \dots \dots (2)$$

Using British units (foot-pound-hour) throughout

$q$  = total heat transmitted in B.Th.U. per hour.

$H$  = heat transmitted in B.Th.U. per square foot per hour.

$L$  = thickness of wall in feet.

$t_1, t_2$  = temperature of hotter and cooler sides of wall in °F.

$k$  = thermal conductivity in B.Th.U. per square foot per hour per °F. per foot thickness (written for the sake of brevity, B.Th.U./sq.ft./hr./°F./ft.)

$A$  = area of surface (square feet) normal to the direction of flow.

Often it is more convenient to express the thickness of the wall,  $L$ , in inches and to take  $k$  as B.Th.U./sq.ft./hr./°F./in. The formula  $H = k(t_1 - t_2)/L$  is still applicable when both of these modifications are applied simultaneously.

### THERMAL CONDUCTIVITIES

The nature of the variation of thermal conductivity of a number of common substances entering into industrial heat problems is shown in Table 36. Both the values for  $k$  just mentioned are given in Table 36.

The difference between metals and non-metals will be evident from Table 36. The thermal conductivity will also be seen to vary with the temperature and may increase or decrease with rise of temperature according to the material. Most metals conduct heat less well with rise of temperature, whereas the conductivity of most refractory materials increases with increasing temperature.

When calculating the rate of heat transmission through a substance as in Fig. 35, the thermal conductivity to be used is the average over the temperature range  $t_1 - t_2$ . The value of the coefficient for aluminium at 400° C. is 1.240 B.Th.U./sq.ft./hr./°F./in., at 600° C. it is 1.050, and over the range 400–600° C. the average may be taken as 1.145.

6 2.000

TABLE 36  
THERMAL CONDUCTIVITY OF SOME COMMON MATERIALS

Material	Temp. °C.	Thermal conductivity *	
		B.Th.U./sq.ft./ hr./° F./in.	B.Th.U./sq.ft./ hr./° F./ft.
Aluminium .. .. .	0	1,400	117
	200	1,490	124
	400	1,730	144
	500	1,860	155
	100-200	2,916	243
Copper .. .. .	100-370	2,700	225
	100-541	2,620	218
	100-837	2,500	208
	18	418	35
Wrought iron .. .. .	100	416	35
	100	314	26
Cast iron .. .. .	100	311	26
Steel .. .. .	100	400	33
Nickel .. .. .	300	366	31
	600	256	21
	800	197	17
	1,200	168	14
	100	203	17
Monel metal .. .. .	400	244	20
	100	711	59
	400	810	68
Firebrick .. .. .	300	7.3	0.61
	700	8.1	0.68
	1,100	8.4	0.70
Magnesite brick .. .. .	300	60	5
	700	37.5	3.1
	1,100	27.5	2.3
Insulating bricks .. .. .	0-200	0.73	0.06
	0-400	0.81	0.07
	0-600	0.90	0.08
	0-800	1.02	0.09
Magnesia pipe covering . . . .	50	0.38	0.03
	100	0.41	0.03
	200	0.46	0.04
	400	0.64	0.05

B.Th.U.

\* The dimensions are strictly ft. hr. ° F., but are stated in the form given for clarity.

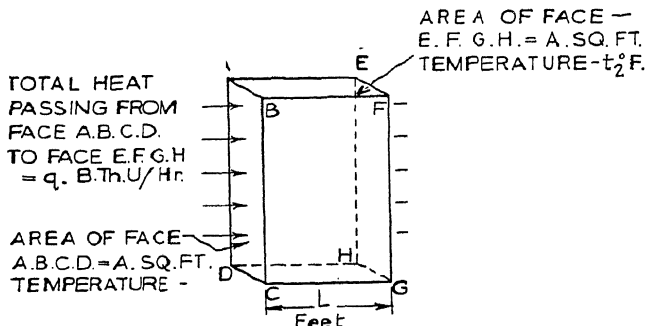


FIG. 35. Heat flow by conduction in a rectangular slab.

refractory wall, 6 inches  
and the wall has been 1  
of the conductivity of  
is known to be 8.2

$$H = \frac{8.2 \times (2,010 - 752)}{6} = 1,200 \text{ B.Th.U./sq.ft./hr.}$$

effect of differences in conductivity, suppose that in  
type, the inner wall is to be maintained at 400° C.  
ired to calculate what must be the temperature of the  
(a) of cast iron 1 inch thick ; and (b) of firebrick  $4\frac{1}{2}$  inches  
of heat transmission  $H = 700 \text{ B.Th.U./sq.ft./hr.}$

Here the amount of heat that is to be passed through the wall is the same in  
both instances, but the thermal conductivity of the materials is very different  
and is taken as 300 for C.I. and 8 for firebrick.

$$\text{For C.I. } 700 = \frac{300 (t_1 - 752)}{1} \text{ whence, } t_1 = 754.3^\circ \text{ F. } (401.3^\circ \text{ C.})$$

$$\text{For firebrick } 700 = \frac{8 (t_1 - 752)}{4.5} \text{ whence } t_1 = 1,146^\circ \text{ F. } (619^\circ \text{ C.})$$

### TRANSMITTANCE OR OVERALL HEAT TRANSFER COEFFICIENT

The calculation of the quantity of heat passing through a wall or a boiler tube lined on one side with soot and some oxide of iron or scale, and on the water side with "boiler scale" requires a knowledge of the surface temperatures of the wall. In practice these temperatures are often not known, but the temperatures of the media on each side of the wall are known. A simple example is a steam pipe. The temperature of the steam inside the pipe, and that of the air surrounding the pipe are known, but not the temperature of the pipe and the insulation. As already explained, to overcome this apparent difficulty it is convenient to use an overall coefficient of heat transfer, obtained in the following manner.

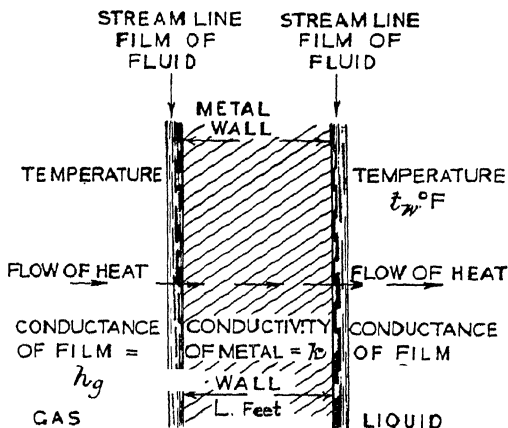


Fig. 36. Heat transmission through a plane wall separating two fluids.

The heat transmission through a simple plane wall separating two fluids and  $w$  is (see Fig. 36) :—

$$= \frac{A(t_g - t_w)}{\quad} \quad \text{per hour} \quad (3)$$

where  $A$  = area of heating surface, square feet.

$t_g$  = temperature of hot medium, °F. (the hot gas in a boiler flue)

$t_w$  = " " " cold " (the water in the boiler)

$h_g$  = surface conductance of hot medium—B.Th.U./sq.ft./hr./°F./ft.

$h_w$  = " " " cold " " B.Th.U./sq.ft./hr./°F./ft.

$L$  = thickness of the wall, feet,

$k$  = thermal conductivity of the wall in B.Th.U./sq.ft./hr./°F./ft.

Defining the transmittance or overall heat transfer coefficient,  $h$ , as follows :—

$$q = Ah(t_g - t_w) \text{ B.Th.U. per hour} \quad (4)$$

and dividing equation (4) into equation (3) and transposing, it is found that

$$h = \frac{1}{\frac{1}{h_g} + \frac{L}{k} + \frac{1}{h_w}} \text{ B.Th.U./sq.ft./hr./°F.} \quad (5)$$

### THE RESISTANCE CONCEPT

The practical significance of these two equations is of supreme importance in heat transmission and is analogous to that of the application of Ohm's law in the transmission of electrical power. In comparison with Ohm's law, which states that the current = voltage/resistance, the temperature potential across the wall ( $t_g - t_w$ ) is equivalent to the electrical potential or voltage and the expression

$$\frac{L}{k}$$

is analogous to the resistance, the total resistance being the sum of the partial resistances. The flow of heat is analogous to the flow of current. Just as the reciprocal of the resistance in an electric wire measures the "conductance" of the wire for electricity and *vice versa*, so the reciprocal of the thermal conductance (e.g.  $1/k_w$ ) measures the resistance to the flow of heat.

It is an important fact that if the partial resistances differ to any great degree in magnitude only the largest resistance will have a significant effect on heat transmission. Thus, in a boiler tube, taking a case quoted by Schack, the surface conductance on the water side,  $h_w$ , is of the magnitude of 1,000, and the corresponding resistance  $\frac{1}{h_w} = 0.001$ . The surface conductance on the flue side,  $h_g$ , is, however, only 6, and resistance = 0.167.

Putting into equation (5) the values of 27.5 for the conductivity of mild steel and a metal thickness of  $\frac{3}{4}$  inch,

$$h = \frac{1}{0.167 + 0.0023 + 0.001} = 5.9$$

virtually equal to the surface conductance on the flue side, which accordingly governs the measure of the heat transmission.

In a condenser surface, on the other hand, conductances on both sides of the heat exchanging surface have high values,  $h_w$  on the water side being approximately 1,000, and  $h_g$  on the steam side being of the order of 2,000 B.Th.U./sq.ft./hr./°F./ft. thickness.

The conductivity of the metal wall can here be shown to be the governing

factor, and reduction of the wall thickness results in an appreciable improvement of heat transmission. It may further be shown \* that, although the thermal conductivity of brass is more than twice that of steel, the transmittance across a 0.2 inch brass wall is only 19 per cent. greater than that across a steel wall of the same thickness. In general the largest partial resistance should first be considered in any attempt to improve heat transmission.

### CONDUCTIVITY FORMULA FOR COMPOSITE WALLS

The standard conductivity formula requires modification when the wall is made of more than one kind of material. If, for example, there are three layers of materials in the wall through each of which the heat flows in turn, the thicknesses respectively being  $L_1$ ,  $L_2$  and  $L_3$ , and the conductivities  $k_1$ ,  $k_2$  and  $k_3$  (Fig. 37) the formula becomes

$$H = \frac{t_1 - t_2}{\frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3}} \quad \text{B.Th.U./sq.ft./hr.} \quad (6)$$

Thus, if a wall consists of 9 inches of firebrick ( $k_1 = 8.1$ ) backed by 3 inches of

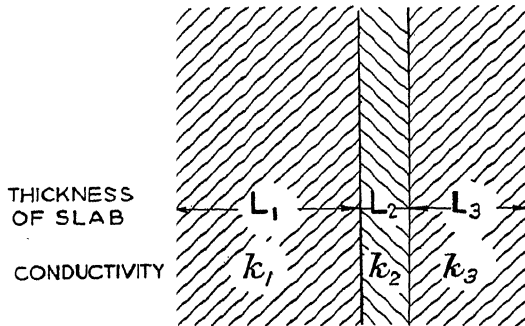


FIG. 37. Conduction through a composite wall.

insulating brick ( $k_2 = 0.8$ ) the inside temperature being  $2,000^\circ \text{F.}$ , and the outer surface  $300^\circ \text{F.}$ , the heat that will pass through is given by:—

$$H = \frac{2,000 - 300}{\frac{9}{8.1} + \frac{3}{0.8}} = 350 \text{ B.Th.U./sq.ft./hr.}$$

A further example applies to the water side of the boiler on which a layer of scale may have formed. Scale has a thermal conductivity of 12–18 B.Th.U./sq.ft./hr./ $^\circ \text{F.}$ /in. thickness, and is thus similar to a firebrick in this respect. Some scales have a much lower conductivity than this. If it is accepted that the safe maximum continuous temperature for boiler tubes is  $900^\circ \text{F.}$ , a calculation of the thickness  $L$  enables an estimate to be made of the thickness of scale that might permit the plates to be dangerously overheated:—

In modern boilers heat input rates may be 70,000–100,000 B.Th.U./sq.ft./hr. If the boiler is at 100 lb. pressure ( $338^\circ \text{F.}$ ) the standard conductivity formula gives

$$70,000 = \frac{\quad}{L}$$

whence  $L = 1/7$  inch.

\* "Industrial Heat Transfer," A. Schack. Trans. by H. Goldschmidt and E. P. Partidge (John Wiley, 1933).

Thus a scale only one-seventh of an inch thickness may set up a dangerous condition in a boiler. The direct effect of scale on thermal efficiency is probably of the order of  $1\frac{1}{2}$  to 6 per cent. loss, according to how much of the total boiler heating surface is coated with scale.

### CONDUCTION THROUGH PIPE WALLS

Although the formulæ for conductivity through a flat surface just given are approximately correct for a curved surface of comparatively large diameter, provided the thickness is small compared with the diameter, they are incorrect when the diameter is small. For calculations of the flow of heat through pipe coverings other formulæ must be used, based on the same general principles.

If the diameter of a pipe is  $d_1$  inches and the diameter of the outer surface of pipe covering is  $d_2$  inches it can be shown that the heat flow is given by:—

$$q = \frac{2\pi k}{2.3 \log \frac{d_2}{d_1}} \cdot (t_1 - t_2) \text{ B.Th.U./hr./ft. run of pipe} \quad (7)$$

where  $t_1$  and  $t_2$  are the temperatures at diameters  $d_1$  and  $d_2$  respectively.

This is perhaps the most useful form for practical purposes, but if the loss of heat per unit surface area of the pipe is required, it is given by:—

$$H = \frac{k}{\frac{d_2}{d_1}} \cdot (t_1 - t_2) \text{ B.Th.U./sq.ft./hr.} \quad (8)$$

where  $r$  is the distance from the pipe axis; when  $r = \frac{1}{2}d_1$ , this gives the heat loss per unit area of the metal pipe, and when  $r = \frac{1}{2}d_2$ , the basis is on the area of the external covering.

Thus if a 6-inch steam pipe is covered by  $2\frac{1}{2}$  inches of magnesia, experiment shows that the surface temperature of the magnesia in reasonably still air is  $105^\circ \text{F.}$  when conveying steam at  $500^\circ \text{F.}$  The thermal conductivity of the magnesia covering may be taken as 0.42 over this temperature range. The heat lost per foot run is then calculated:—

$$\frac{2(500 - 105)}{\log \frac{5\frac{1}{2}}{3}}$$

The heat lost per square foot of surface area of the metal pipe is given by:—

$$\frac{5)}{\log \frac{5\frac{1}{2}}{3}}$$

For the more complete solution of such problems, taking into account the surface conductance inside and outside the pipe, the heat transmission per foot run of composite pipe wall is given by:—

$$q = \frac{\pi(t_s - t_a)}{\frac{1}{d_1 h_s} + \frac{1.151}{k_1} \log \frac{d_2}{d_1} + \frac{1.151}{k_2} \log \frac{d_3}{d_2} + \frac{1}{d_3 h_a}} \text{ B.Th.U./hr./ft. run} \quad (9)$$

where  $t_s$ ,  $t_a$  are the temperatures of the steam and the surrounding atmosphere.  
 $d_1$  and  $d_2$  = internal and external diameters of the metal pipe, feet.

$d_2$  and  $d_3$  = " " " " " lagging, feet.

$k_1$  and  $k_2$  = thermal conductivities of the metal of the pipe and the lagging material.

$h_s$  and  $h_a$  = surface conductance of the internal pipe surface and external surface of the lagging.

The external conductance  $h_a$ , is in this expression made up of the convection and radiation coefficients,  $h_c$  and  $h_r$ , explained later.

### CONVECTION

Convection is dependent upon the laws of fluid flow (Chapter IX). Osborne Reynolds showed the difference between streamline or laminar flow and turbulent flow by introducing a fine stream of a red liquid into water flowing through a glass tube. Fluid friction increased instantaneously with the transition from streamline to turbulent motion, and the law governing the pressure drop through the tube changed. With streamline flow it was directly proportional to the quantity of fluid flowing through the tube, with turbulent flow to the square of this quantity.

In natural convection the motion of the fluid is caused by gravity due to differences in density between hotter and cooler parts; in forced convection the fluid motion is caused by external forces, such as a chimney draught or a fan. In industrial work, the motion induced by a chimney is sometimes regarded as a form of natural convection, the term forced convection being reserved for motion set up by fans or other external means.

A hot surface, e.g. a steam pipe or a heated tank, over which the air flows freely, is losing heat to the air by natural convection. A nest of tubes in a condenser over which gas is blown by a blower or fan is absorbing heat from the gases by forced convection.

Progress in the study of convection in its bearing on heat transfer has been made by the application to the problem of dimensional analysis and the theory of similarity, a means whereby data obtained from one set of observations can be applied to another provided they are dimensionally compatible and bear geometrical similarity to one another. Thus results obtained from models can be applied to industrial plant. Further the method has been the instrument of correlating and establishing the validity of formulæ determined on widely differing types of apparatus, as well as the means of providing a technique for solving individual types of problems. For further information reference should be made to the modern standard works.

### BASIS OF STUDY OF CONVECTION—FLOW IN PIPES

Reynolds showed in his studies of the motion of water \* that certain relationships existed in regard to the velocity ( $v$ ), the depth of the fluid stream ( $D$ ), the density of the fluid ( $\rho$ ) and the viscosity ( $\eta$ ), which could be grouped together to give a criterion known as Reynolds's Number ( $Re$ ), a fuller account of which will be found in Chapter IX. It is sufficient to state here that

where  $v$  and  $\rho$  = velocity and density at the conditions of temperature and pressure ruling, and the viscosity,  $\eta$ , is defined in various ways using either unit force or unit mass. The value of the criterion  $Re$  is the same whatever consistent system of units be used. The change from streamline to turbulent flow is associated with a value of  $Re$  = about 2,300. It has been further shown that the characteristics of flow are associated with the heat transference from moving fluids, and that resistance to fluid flow and heat transfer are closely related (see Chapter IX).

The velocity distribution in two streams of fluids flowing in pipes will be similar if the Reynolds numbers are the same for both. The temperature distribution in the streams will be the same if the expression  $vD/a$ , where  $a$  is the thermal diffusivity, is the same for both; further, for two fluid streams to have similar temperatures, the expression  $hD/k$  must be the same for both where the diffusivity =  $k/c\rho$

\* Ref. "On the Motion of Water." Trans. Roy. Soc., London, 174, 935 (1885).

$k$  = thermal conductivity  
 $c$  = sp. ht. per unit weight  
 $\rho$  = density  
 $h$  = surface conductance  
 $D$  = width of stream

Study of these groups has shown that for specific applications they are related by some simple value. Thus Nusselt developed the relationship shown by the equation

the dimensionless group  $\frac{hD}{k}$  being known as the Nusselt number  $Nu$ , and he found for heat transfer in a brass tube of inside diameter 0.87 inch that the surface conductance,  $h$ , increased with the 0.786 power of the velocity,  $V$ . That is to say

$$h = C \cdot V^{0.786} \quad (11)$$

where  $C$  represents all the variables unchanged in the experiments.

It can be shown by the development of the above equations that the surface conductance is derivable from

$$\frac{hD}{k} = C' \cdot V^{0.786} \quad (12)$$

in which  $C'$  is a general constant determined by the specific conditions. Thus for forced turbulent flow of gas in pipes

$$\frac{hD}{k} = C' \cdot V^{0.786} \quad \text{B.Th.U./sq.ft./hr./}^\circ\text{F.} \quad (13)$$

Certain transformations have been made and the values are

$k_{\text{wall}}$  = thermal conductivity of the gas at the temperature of the pipe wall, B.Th.U./sq.ft./hr./ $^\circ\text{F./ft.}$

$k$  = thermal conductivity of the gas at its mean temperature, B.Th.U./sq.ft./hr./ $^\circ\text{F./ft.}$

$C_p$  = specific heat of the gas under the conditions of flow, B.Th.U./cu.ft./ $^\circ\text{F.}$

$V$  = velocity of the gas flow, feet per second.

$D$  = internal diameter of the pipe, feet.

It is also possible to derive expressions for the calculation of  $h$  by the use of the Reynolds' criterion, since the value of this is constant for similar streams. Generally the subject is complex and beyond the present aim to provide a basis for study and indicate the sources of the most essential formulæ. Simplifying the expression by applying the normal values of the constant for chimney gases and air the surface conductance by convection for turbulent flow in long smooth pipes at any pressure is given by

$$h = \frac{0.32 v_o^{0.8}}{D^{0.2}} \text{ B.Th.U./sq.ft./hr./}^\circ\text{F.} \quad (14)$$

where  $v_o$  = fluid velocity in feet per second calculated to S.T.P.

$D$  = diameter of the duct, feet.

For ducts which are not circular or square, the equivalent diameter is given by  
 (area of cross-section  $\times 4$ )/perimeter.

For further reference to the literature of a wide field of experimental investigation in this class of problem which has been the basis of the useful empirical formulæ given in the remainder of this discussion, the reader should consult



more specialised works on the subject, e.g. McAdams' "Heat Transmission" (2nd Ed., 1942), Fishenden and Saunders' "The Calculation of Heat Transmission" and Schack's "Industrial Heat Transfer" (trans. by Goldschmidt and Partridge).

### NATURAL CONVECTION

The transfer of heat by natural convection is governed by the area of the surface, the shape and position of the surface and the temperature difference between the surface and the gas. A useful empirical expression is,

$$H = C (t_1 - t_2)^{1.25} \text{ B.Th.U./sq.ft./hr.} \quad (15)$$

where  $t_1$  is the temperature of the surface,  $t_2$  of the gas or air, and  $C$  is a constant depending on the shape and position of the surface.

The value of  $C$  as determined by experiment is as follows:—

$C = 0.39$  for a plane horizontal surface facing up and hotter than the gas; or facing down and colder than the gas.

$C = 0.3$ , for plane vertical surfaces and for large bodies of irregular shape but without re-entrant angles.

$C = 0.2$  for horizontal surfaces facing downwards and hotter than the gas.

For large horizontal cylinders (and this applies to pipes) over 6 inches diameter  $C = 0.35$ . As the cylinder diameter decreases (e.g. with small pipes) the value of  $C$  increases rapidly, so that for a wire of 0.01 inch diameter  $C = 3.5$ .

Some idea of the magnitude of natural convection effects can be seen from Fig. 38.

As a simple example of the use of convection formulæ, consider a bare hot water tank nearly filled with hot water at  $180^\circ \text{F}$ . placed in a workshop with air

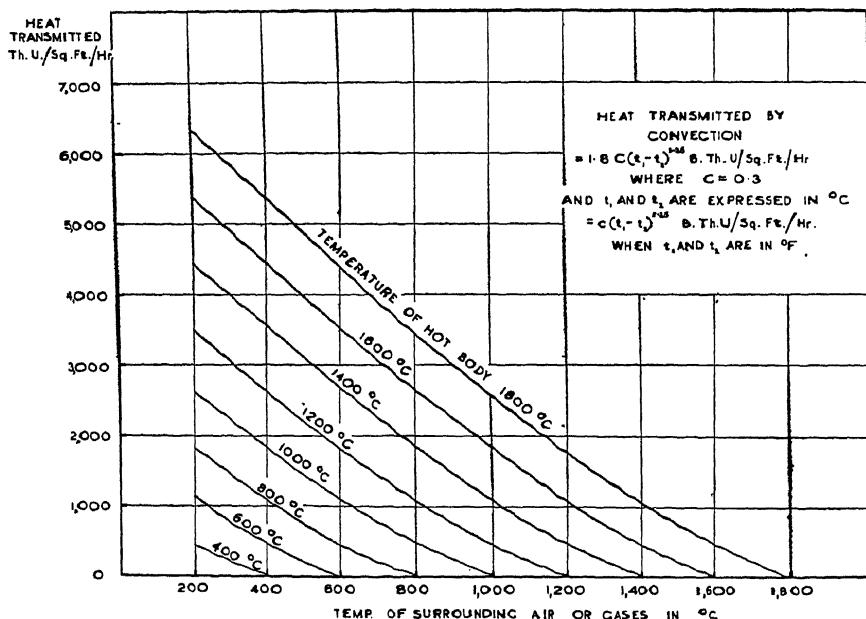


FIG. 38. Heat transfer by natural convection in still air.

temperature of 70° F. The tank is supported on four columns so that its base is exposed. It measures 20 feet  $\times$  14 feet wide, and is 10 feet high. It is connected to an apparatus where the water is used by a pipe 8 inches diameter and 15 feet long. It is required to calculate the loss of heat per hour by convection.

Area of connecting pipe	..	..	..	sq. ft.
Area of top of the tank	..	..	..	31.4
Area of bottom of the tank	..	..	..	280
Area of sides of the tank	..	..	..	280
	..	..	..	680
Heat loss from connecting pipe = $0.35 \times 31.4 \times (180 - 70)^{1.25}$				
= 3,900 B.Th.U. per hour.				
Heat loss from top of tank = $0.39 \times 280 \times (180 - 70)^{1.25}$				
= 40,000 B.Th.U. per hour.				
Heat loss from bottom of tank = $0.2 \times 280 \times (180 - 70)^{1.25}$				
= 20,000 B.Th.U. per hour.				
Heat loss from sides of tank = $0.3 \times 680 \times (180 - 70)^{1.25}$				
= 72,600 B.Th.U. per hour.				

$$H = 3,900 + 40,000 + 20,000 + 72,600 = 136,500 \text{ B.Th.U. per hour.}$$

In addition to this there is loss by radiation which will be dealt with in a later section. Since the tank is unlagged it has been assumed in the foregoing example that the surfaces of the tank are at the same temperature as that of the water. The loss of heat can be reduced if the value of the temperature drop  $(180 - 70)^{\circ} \text{F.}$  can be decreased. If the tank and pipe are lagged, the surface temperature should be reducible at least to 90° F. The loss of heat by convection will then be reduced to 16,200 B.Th.U. per hour—a striking commentary on the value of adequate insulation.

## FORCED CONVECTION

In describing earlier the basis of the study of convection some indication of

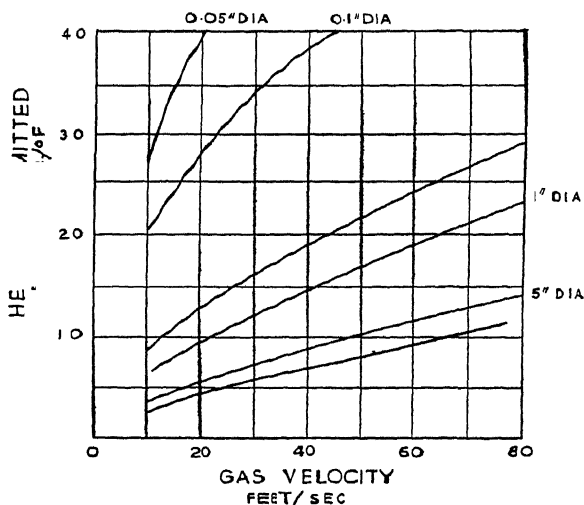


FIG. 39. Heat lost from cylinders by forced convection to air flowing at right angles to the cylinder.

(Based on "The Calculation of Heat Transmission," by M. Fishenden and O. A. Saunders, H.M. Stationery Office.

the most useful formula has already been given, from which it will be noted that the rate of heat transfer depends directly upon the difference of temperature between the hot and cold elements of the heat exchanging surface, and not upon the  $\frac{5}{4}$  power of the temperature difference, i.e.

$$H = h (t_1 - t_2) \quad (16)$$

where  $h$  is the "surface conductance," and is analogous to  $k$ , the thermal conductivity in the simplest case of conduction through a solid slab. The parallel is, however, far from being exact, for unlike  $k$ , which is a characteristic of the material, the value of  $h$  depends upon the linear velocity of the gas stream, the size of the body, and the physical properties of the gas.

The above formula is applicable to a wide range of conditions, the value of the surface conductance naturally varying with circumstances.

The following statement defines the values in certain specific applications.

*Single tube in a transverse stream of air.*

Fig. 39 shows the values found for  $h$  for tubes of various sizes. If the turbulence of the gas stream is artificially increased these values may be increased by up to 50 per cent. Increased turbulence means increased resistance, and therefore increased draught is needed, so that this method of increasing the heat flow must be used with caution. A Galloway tube is an example of this arrangement.

*Single tube in longitudinal flow of air.*

The mean value of  $h$  for any given conditions is about half that for the transverse stream, but can be doubled by creating artificial turbulence.

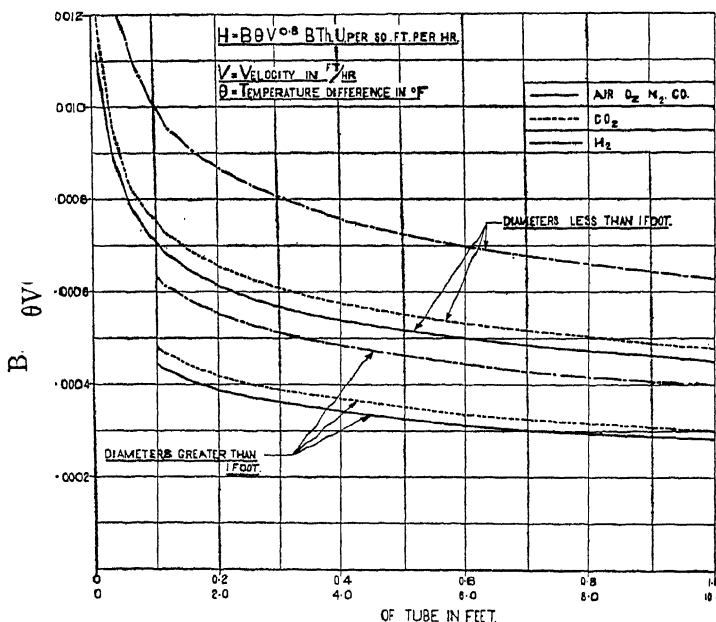


FIG. 40. Coefficients of forced convection for air or other gases flowing through tubes of varying diameter.

(From "The Calculation of Heat Transmission," by M. Fishenden and O. A. Saunders, H.M. Stationery Office.)

*A bundle of tubes in a transverse stream of air.*

This construction is that of an economiser or condenser. A staggered arrangement of the tubes through creating greater resistance is better than a straight-through flow.

*The use of fins.*

In air heaters, and other plant, the heat transfer by forced convection is increased by adding fins or other projections to increase the effective surface area. The heat transfer to fins is actually less than the heat transfer to the bare tube, but in view of the increased exposed surface, the overall heat transfer referred to the bare tube surface is increased. There is a limit to the useful height of the fins and their spacing. The larger and more numerous the fins the greater will be the heat transfer for a given base area; but increase in height causes a reduction in the mean temperature of the exposed surface and if the fins are too close the flow of air between them is impeded. The fins must be of sufficient thickness to conduct the heat to or from the solid plate without too great a temperature gradient.

*Flow through a single tube*

The circular flues of a shell boiler, a main carrying hot producer gas, some forms of air heater, or any other arrangement in which the gas being heated flows through a tube, are examples of this class of problem. Here it is found that

$$H = B v^{0.8} (t_1 - t_2) \quad \text{B.Th.U. per square foot per hour} \quad (17)$$

$v$  is the velocity of the gases or liquids passing through the tube in feet per hour. The value of  $B$  is given in Fig. 40. Another treatment of this arrangement on more general lines is that already described, equation (14).

**HEAT TRANSFER AND PRESSURE DROP IN FLOW ACROSS BANKS OF TUBES**

In discussing the contribution made by Osborne Reynolds to our knowledge of fluid flow, the importance of the resistance to flow introduced by fluid friction was indicated. In practical design, heat transfer and the resistance to flow through the appliance must always be considered together. Since the cost of equipment and the cost of power tend in opposite directions, there is an economic balance for some optimum value of resistance to fluid flow and the area of the heating surface.

In banks of tubes, occurring in so many forms of heating appliances and exchangers, the mean rate of heat transfer  $H$  per unit tube surface is given in all cases of forced convection by a relation between the Nusselt number (a dimensionless group,  $\frac{Hd}{k\theta}$ ) and the Reynolds' number,  $Re$ , which depends upon the geometrical arrangement of the tubes.

The relation, which fits the most reliable experimental data is

$$\frac{Hd}{k\theta} = 0.29 \quad \frac{10.61}{\eta} \quad (18)$$

$H$  = rate of heat transfer per unit area, B.Th.U./sq.ft./hr.

$d$  = diameter of pipes, feet.

$k$  = thermal conductivity of fluid.

$\theta$  = temperature difference between fluid and surface, °F.

$v d \rho / \eta$  = Reynolds' number as previously described.

and the constant  $C_{\eta}$  depends upon the tube arrangement. The velocity  $v$  is measured at the narrowest restriction. This may occur between adjacent tubes in line, or between two diagonally opposed tubes.

Frictional resistance to flow is given by the well-known D'Arcy formula (also known as the Fanning equation), which is discussed in detail in Chapter IX. According to this expression, if

$$\begin{aligned} h &= \text{pressure drop in inches w.g.} \\ f &= \text{coefficient of friction.} \\ g &= \text{gravitational constant} \\ D &= \text{pipe diameter (feet)} \\ \rho &= \text{gas density in lb. per cubic foot.} \\ v &= \text{velocity in feet per second.} \end{aligned}$$

$$\frac{2f\rho v}{gD} \quad (19)$$

For flow through tubes this can be resolved into

$$h = 10^{-3} C_F n \rho v^2$$

where  $C_F$  is a coefficient of discharge depending on the tube arrangement and  $n$  is the number of rows of tubes.

Values of the constants  $C_H$  and  $C_F$  for various arrangements of tube banks, both in line and staggered are to be found in a recent paper by Lander.\* The magnitude of the heat transfer is proportional to the corresponding value of the coefficient  $C_H$  in both in-line and staggered arrangements. For "in-line" arrangements with values of the Reynolds number above 20,000,  $C_H$  becomes unity for all kinds of spacings, from which it follows that the heat transfer is determined by the velocity. For staggered tubes it drops slightly. The mean value of the heat transfer at wide spacings in line is some 38 per cent. below that for a single tube, due to the shielding effect of one tube on another. For staggered formations the pressure drop appears to be quite independent of the longitudinal space, but rises as the lateral spacing is decreased.

In general, a heat exchanger becomes more economical as the pipe diameter is decreased. The installation cost decreases approximately with the fourth root of the pipe diameter, but practical conditions limit the reduction of cross section. Where heating surface is relatively expensive it will not be economical to use stack draught, since the velocity would probably be too far from the optimum value.

## HEAT TRANSFER TO SINGLE PIPES

The best simple equation for highly turbulent flow of air against pipes is given by\* :—

$$\frac{v_o^{0.56}}{D^{0.44}} = \dots \dots \dots (21)$$

$v_o$  is the velocity in feet per second at N.T.P.;  $D$  is the pipe diameter in feet.

For undisturbed flow the constant becomes 0.70.

## PIPE BUNDLES

A practical equation for conductances by convection from air to rows of evenly arranged or staggered pipe bundles has the following general form\* :—

$$\frac{C \cdot v_o^x}{D^y}$$

$v_o$  is measured at N.T.P. as in (21).

The values for the various groupings are given in Table 37. They are valid for air at any temperature and pressure, and are approximately valid for flue gases.

\* "A Review of Recent Progress in Heat Transfer," C. H. Lander, Journal, Institution of Mechanical Engineers, April 24th, 1943, pp. 95-98.

TABLE 37

Arrangement of pipe bundles		Even rows	Staggered rows
Values of indices	x	0.654	0.69
	y	0.346	0.31
No. of rows		C	C
2		0.615	0.70
3		0.635	0.79
4		0.65	0.86
5		0.66	0.91

### HEAT TRANSFER FROM FLUIDS TO WALLS

In forced convection in a plane wall, as distinct from natural convection, the heat transfer has been found experimentally to depend not only upon whether the plate or wall is vertical or horizontal, but also on the character, particularly the roughness, of its surface. Data based mainly on experiments by Jürges is as follows (Table 38).

TABLE 38  
FORCED CONVECTION IN PLANE WALLS  
(Values of  $h$  in B.Th.U./sq.ft./hr./°F.)

For velocities smaller than 16.5 ft./sec.

For velocities larger than 16.5 ft./sec.

$h_{cs} = 0.98 + 0.20 v$ , for a smooth surface  
 $h_{cr} = 1.02 + 0.20 v$ , for a rolled surface  
 $h_{cr} = 1.09 + 0.22 v$ , for a rough surface

$h_{cs} = 0.50 v^{0.78}$   
 $h_{cr} = 0.50 v^{0.78}$   
 $h_{cr} = 0.5$

The air velocity  $v$  is here referred to a basis of 70° F. If the temperature varies much from 70° F. the correction should be applied to bring the corresponding value of  $v$  to 70° F. Thus if the temperature is 200° F., and the actual velocity 20 feet per second, the corrected value of  $v$  becomes

$$v = 20 \times \frac{460 + 70}{460 + 200} = 16.05 \text{ feet per second.}$$

For surfaces greater than  $1\frac{1}{2}$  square feet  $h$  becomes decreased by as much as 40 per cent. for surfaces several feet in length.

### HEAT TRANSFER FROM SUPERHEATED STEAM WITHIN PIPES

Whilst the fundamental investigations of heat transmission by convection have produced equations for practical use which are applicable to all fluids, condensing and boiling liquids introduce special conditions. It is doubted by some authorities if the equations for superheated steam are sufficiently satisfactory, and further investigation is probably needed. Schack gives the best available general equation for superheated steam in pipes, as follows:—

$$h = \left( 0.432 + 0.151 \frac{1,000}{D^{0.16} L^{0.05}} \right)^{0.78} \text{ B.Th.U./sq.ft./hr./°F.} \quad (23)$$

The inclusion of a term involving the length of the pipe is on account of decreased turbulence in the fluid flow with increase in pipe length, the value of the exponent 0.05 having been established by Nusselt.

## CONVECTION IN LIQUIDS

It has been shown in reference to the work of Osborne Reynolds that the viscosity of a fluid is linked with the mechanics of its flow (see Chapter IX, Fig. 56). The layer of fluid in contact with the surface is virtually stationary; the next layer flows over this stagnant layer with an imperceptible velocity, and so on, each succeeding layer moving with a higher speed. The greater the viscosity of the fluid the more is the resistance to motion.

The temperature drop through the film has been illustrated in Fig. 34. The significance of the velocity in influencing the thickness of the non-conducting stationary layer and so the surface conductance, and the reason for the use of the alternative term "film coefficient" is thus readily apparent. It is accordingly not surprising to find that for water flowing slowly in pipes the conductance has a remarkably low value. In practice the surface conductance from or to bodies in contact with water has values of the order of 40–200 according to the opportunity for the development of convection currents.

Fluid flow arising from the boiling of a liquid at a heating surface must be distinguished from free convection, since the ebullition introduces a measure of forced convection. The surface conductance for heat transmission to boiling water is of the order of 1,000–4,000 B.Th.U./sq.ft./hr./°F. according to the intensity of boiling and conditions of water circulation. The aggregation of steam bubbles over an intensely heated area resulting in overheating owing to the insulating effect of the layer of gas bubbles is a recognised occurrence in regions of poor circulation. The fact that the heat transfer is somewhat greater for vertical than for horizontal surfaces is thus also to be expected. It has also been established that the more viscous the liquid the lower is the surface conductance under conditions of ebullition.

The correlation of experimental results on the basis of similarity transfer by forced convection for liquids flowing in tubes in turbulent motion has been shown by Fishenden and Saunders (*loc. cit.*, p. 186) to be

$$Hd \left( \frac{c\nu}{k} \right)^{0.5} = 0.03 \left( \frac{vcd}{\nu} \right)^{0.8} \quad (24)$$

where  $H$  = heat flow per unit area per unit time, B.Th.U./sq.ft./hr.

$d$  = diameter, feet.

$k$  = thermal conductivity, B.Th.U./sq.ft./hr./°F./ft.

$\theta$  = temperature difference, °F.

$c$  = specific heat per unit volume at constant pressure, B.Th.U./cu.ft./°F.

$\nu$  = kinematic viscosity, sq.ft./sec. =  $\eta/\rho$  (see Chapter IX).

$v$  = linear velocity, feet per second.

all values being in consistent units, either English or metric.

A simplified equation of the surface conductance of water with temperature  $t_w$  and with water flowing in pipes at an actual velocity of  $v$  feet per second, based on a density of 62.3 lb. per cubic foot is given by

$$H = 150 (1 + 0.011 t_w) v^{0.8} / d^{0.2} \quad (25)$$

where  $d$  = internal pipe diameter in inches and  $t_w$  is in °F.

Thus a flow of water of 9,000 lb. per hour passing through a pipe of 3 inches internal diameter will give a surface conductance at 200° F. of 328 B.Th.U./sq.ft./hr./°F. The drop of temperature through such a pipe freely exposed to the atmosphere is found under these conditions to be roughly of the order of 1° F. Further, this rate of loss of heat is just equal to the rate of loss of heat from a bare pipe by radiation and convection when there is an atmospheric temperature of 60° F. By calculating the drop of temperature in the water from a





## LIQUID CONVECTION CURRENTS—BOILER CIRCULATION

Since water is a very poor conductor of heat, the absorption of the heat transmitted from the fire to the water side of boiler heating surface must be effected by currents within the body of the liquid. In comparatively stagnant water, experiments tend to prove that the water film adjacent to the heated metal is heated by conduction, and subsequently breaks up into numerous small steam bubbles. These, unless removed by rising through the liquid or by the flow of water over the surface to which they are attached, form a heat-resisting blanket, and prevent the dispersal of heat by eddy currents throughout the main volume of water. This effect may be observed in boilers having water "legs" or regions of poor circulation, the aggregation of steam bubbles over an intensely heated area resulting in overheating and possibly deformation of the plate. Water circulation is therefore of vital importance for even moderate rates of heat transmission.

Unless otherwise constrained, steam bubbles tend to rise in a vertical direction, and since the greatest number is evolved in the region of the greatest heat transmission, the heating surface must be designed with free and adequate circulation on the water side. In an internally-fired boiler, for instance, the greater evaporation takes place over the furnaces. The water space above the furnaces, therefore, should be clear of obstructions, and when tube nests are provided the tubes must be as widely spaced as possible.

In practice it is usual to provide suitable "lanes" between tube nests, through which the ascending hot water and steam bubbles flow once circulation is established. An upward-moving current implies the existence of a downward flow to replace the water displaced, and similar provision must be made in this respect. In Lancashire boilers the colder water may flow down the space between the flues and shell, while in horizontal multitubular boilers a space of at least 3 inches must be left for this purpose between the outer tube peripheries and the boiler shell.

Neglect of adequate circulation lanes in shell boilers will result in erratic and violent ebullition, and when regions of the water space do not participate in the general circulation, heavy thermal stresses may be set up in the adjacent metal parts. This sometimes occurs in badly-designed boilers of the internally-fired type.

In water-tube boilers the necessity and conditions for good circulation on the water side are the same, but their achievement in practice is complicated by the high rates of heat transmission possible on this type of boiler. Natural circulation is a function of the ratio between the density of steam and that of water, or, in other words, the natural circulation of water implies the existence of a difference in temperature in the water spaces of the boiler. The more rapid the circulation the greater is the tendency for the temperature to be equalised throughout the boiler, so that each increase in rapidity of circulation reduces the force producing the circulation.

The trouble is further aggravated by the modern use of high feed water temperature and sometimes complicated water-wall system. Difficulty has also been experienced at times in arranging for an adequate flow of water to all the water tubes, the high rate of flow in the tubes nearest the fire tending to starve those tubes at the colder end of the boiler heating surface. Conditions may also arise in intermediate tubes which lead to a temporary stoppage or even a reversal of flow.

Some years ago it was thought that these and similar disadvantages would militate against the use of natural circulation in water-tube boilers for very high pressures and ratings, but their effect has been practically eliminated by careful design in the distribution of water flow to the various tube banks

and the provision of suitable downcomer surface. As a result, natural circulation in boilers is now considered to be suitable for steam pressures up to 2,500 lb. per square inch. Moreover, the design of modern high-pressure, high-duty boilers is such that the actual boiler heating surface is used mainly for steam generation and superheating; the sensible heat stage is effected by the use of steam bled from the main turbines to heat the feed water, and by the provision of separate convective surface for feed water and air heating. Even when the type of water-tube boiler employed necessitates the provision of boiler surface in regions of low gas temperatures, the gas temperature difference between the radiant and the convective zones is such that circulation troubles are largely smoothed out.

### RADIATION

It has been stated that a hot solid body sends out radiation over a wide range of wave lengths. The manner in which the intensity of radiation varies with wave length and with the temperature of the hot body has an important bearing on heating practice. The most important information required, however, is the total energy that is transmitted to take part in the heating process.

A further significant circumstance arises also in regard to the capacity of a body to absorb radiation. A perfect absorber is known technically as a "black body." A perfect black body is said to have an "emissivity" of 1. "Black body" conditions are approached in a closed furnace chamber, where any radiation at first not absorbed is repeatedly reflected until it is finally absorbed. Even so good an absorber as lamp black is not a perfect black body; it absorbs to the extent of 0.97 of the black body absorption.

A good absorber is also a good radiator. Thus if lamp black, which is equivalent to particles of carbon in a flame, is heated, it will radiate to the extent of 0.97 of the maximum amount theoretically possible. Many bodies such as polished metals are poor absorbers and accordingly also poor radiators, their emissive power being below 0.1 of that of a "black body."

Stefan found experimentally that the total radiant energy emitted by a hot body varied as the fourth power of the absolute temperature of the body. Boltzmann developed the theoretical basis, and the result was the Stefan-Boltzmann law.

The Stefan-Boltzmann law states that when a body of area  $A$  square feet and having an emissivity,  $E$ , is at  $T_1^\circ$  Abs. ( $= t + 460^\circ$  F.) and its solid surroundings are at  $T_2^\circ$  Abs., the heat radiated,  $Q$ , is given by

$$Q = 17.3 \times 10^{-10} \times EA(T_1^4 - T_2^4) \text{ B.Th.U. per hour} \quad (27)$$

A form convenient for calculation is

$$Q = 0.173 EA \left[ \left( \frac{T_1}{100} \right)^4 - \left( \frac{T_2}{100} \right)^4 \right] \quad (27a)$$

### DEFINITION OF SURFACE "A" AND THE CONSTANT "E"

If radiation takes place between two surfaces, one of which is totally enclosed by the other, then the smaller or enclosed surface is referred to as the area  $A$ , regardless of the direction in which heat is being radiated. Thus in a combustion chamber containing a fuel bed the area of the grate rather than that of the walls of the chamber is implied.

### EFFECTIVE SURFACE AREA

The Stefan-Boltzmann radiation formula has been expressed in terms of B.Th.U. radiated (or absorbed) per square foot of area per hour. The area of the hotter and cooler surfaces involved may not be the same, and generally will not be the same. It is important to note that invariably the expression "rate

of heat transfer per square foot " applies to the area of the enclosed (i.e. the smaller) surface.

The heat transfer per square foot of the larger surface may be found by calculating the total heat exchange from the area of the smaller surface and dividing this by the area of the larger surface.

For a surface having no re-entrant angles, the total area, whether plane or curved, is the effective radiating surface.

For a surface having re-entrant angles, such as irregular-shaped stock in an annealing furnace, or many furnace structures, the increased area due to the concavity must not be included in the total area; the area that must be taken is defined as the least surface that could be built across the concave portion.

The net radiation between two surfaces is dependent not only upon the radiation constant of the radiating surface, but also upon that of the surface receiving radiation. For two parallel mutually radiating surfaces, having constants respectively  $E_1$  for the hotter and  $E_2$  for the cooler, the net radiation constant is

$$E = \frac{E_1 E_2}{E_1 + E_2 - E_1 E_2} \quad (28)$$

The problem becomes more complicated in practice because three other laws have to be borne in mind, namely, Kirchoff's and Lambert's laws and the law of inverse squares.

In addition, the relative sizes of the surfaces influence the extent of radiation. Equation (28) gives the minimum value for the emissivity; the maximum value is  $E_1$ . The maximum value represents the heat loss from a hot surface very small compared with its surroundings, under which condition the emissivity of the (cooler) surroundings  $E_2$  has practically no effect on the heat loss. The minimum value represents the heat loss from a substance nearly as big as its surroundings.

### KIRCHOFF'S RADIATION LAW

Kirchoff's law states that the radiating capacity of a given body, represented by the radiation constant  $E$ , is for a given temperature and a given wave length proportional to the absorbing capacity of the body. In other words a body at a certain temperature will radiate in proportion to its ability to absorb. Thus polished metals absorb only a small percentage of the radiation falling upon them and have a small relative blackness, whereas lamp black being nearly a perfect radiator approaches being a perfect absorber.

### LAMBERT'S LAW

The radiation law of Lambert states that the radiation from a surface in a

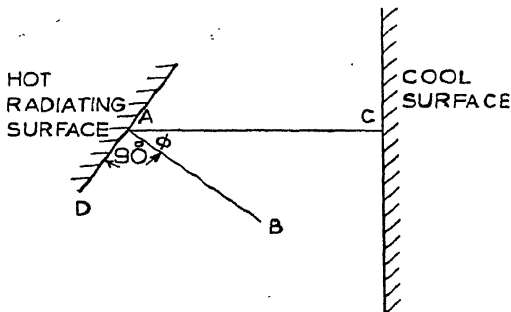


FIG. 41. Lambert's law of radiation.

direction at an angle with the surface varies as the cosine of the angle between the direction of radiation and the normal to the surface (Fig. 41). The radiation in the given direction is

(29)

where  $q_n$  is the radiation normal to the surface.

### INVERSE SQUARE LAW

The intensity of radiation from a point source of radiant energy decreases with the square of the distance from the point source.

Another difficulty in the computation of radiation is that some bodies exhibit relative absorption of certain wave lengths.

For the full treatment of a somewhat complicated subject reference must be made to standard works on heat transmission. Actually in practice calculations on the basis of equation 27, using the emissivity constants given in Table 40 serve a useful purpose in many problems without recourse to the more exact treatment referred to above.

TABLE 40. EMISSIVITY OF SOLID SURFACES

Material	Surface conditions	Emissivity (Black body = 1)
Brass	Brightly polished .. .. .	0.057
	Recently rubbed with emery .. .. .	0.208
Copper	Polished .. .. .	0.041
	Etched or scratched .. .. .	0.094
	Oxidised black .. .. .	0.788
Aluminium	Polished .. .. .	0.053
	Rough .. .. .	0.072
Lead	Grey oxidised .. .. .	0.284
Iron	Polished nickel-plated .. .. .	0.059
	Matt nickel-plated .. .. .	0.114
	Fresh tinned .. .. .	0.082
	Fresh zincd .. .. .	0.230
	Grey zincd .. .. .	0.280
	Freshly rubbed with emery .. .. .	0.245
	Red rusted .. .. .	0.694
	Wrought .. .. .	0.664
	Coarse oxide .. .. .	0.817
	Cast .. .. .	0.817
Varnished enamel	Snow-white .. .. .	0.919
Aluminium	Varnished .. .. .	0.401
Paper	.. .. .	0.940
Plaster of Paris	$\frac{1}{8}$ mm. thick .. .. .	0.915
Oak wood	Planed .. .. .	0.941
Brick	Red .. .. .	0.941
Porcelain	Glazed .. .. .	0.936
Glass	Polished .. .. .	0.950
Marble	Light grey, polished .. .. .	0.943
Aluminium paint	On rough sheet iron .. .. .	0.38-0.7
		average 0.55
Metallic paints	All colours .. .. .	0.9-0.95
Furnace interior	Closed chambers at uniform temperature—all surfaces .. .. .	1
Refractory brick	Firebrick .. .. .	0.75-0.8

The lower the value of the emissivity  $E$ , the less is the heat lost from a hot body, or the heat gained by a cool body. If under black body conditions the temperatures are such that  $H = 100$  B.Th.U. per square foot per hour, the heat lost from a cast-iron surface under the same conditions would be 81.7

B.Th.U. per square foot per hour, and that lost from a surface of polished copper would be no more than 4.1 B.Th.U. per square foot per hour.

The experimental fact that the colour of a surface does not necessarily affect its emissivity often leads to confusion and seems contrary to known experience that in hot countries white clothes are found to be cool and dark clothes are uncomfortably hot. The explanation lies in the fact that the emissivity (and therefore the absorptivity) of substances varies with the wave length of the radiation. The low temperature absorptive power given in Table 21 often differs very considerably from the emissivity or absorptivity to solar radiation. Whereas terrestrial sources of radiation are generally between atmospheric temperature and 1,600° C., the solar surface is over 5,000° C.

### EFFECT OF TEMPERATURE ON RADIATION

From the Stefan-Boltzmann law it will be seen that the rate of heat transmission by radiation varies as the fourth power of the absolute temperatures. Conduction, on the other hand, varies as the first power, and convection with the 1.25th power. It is thus evident that the higher the temperature difference between the hotter and cooler surfaces, the more important relatively is the transference of heat by radiation.

It is thus an established principle that for rapid transference of heat by radiation, the temperature of the radiating surface should be as high as possible. An application of this principle is the use of preheated air to increase flame temperatures (cf. Fig. 19, Chapter V).

To make this clear by a numerical example, let it be supposed that heat is to be transferred by radiation from a surface at high temperature to a surface

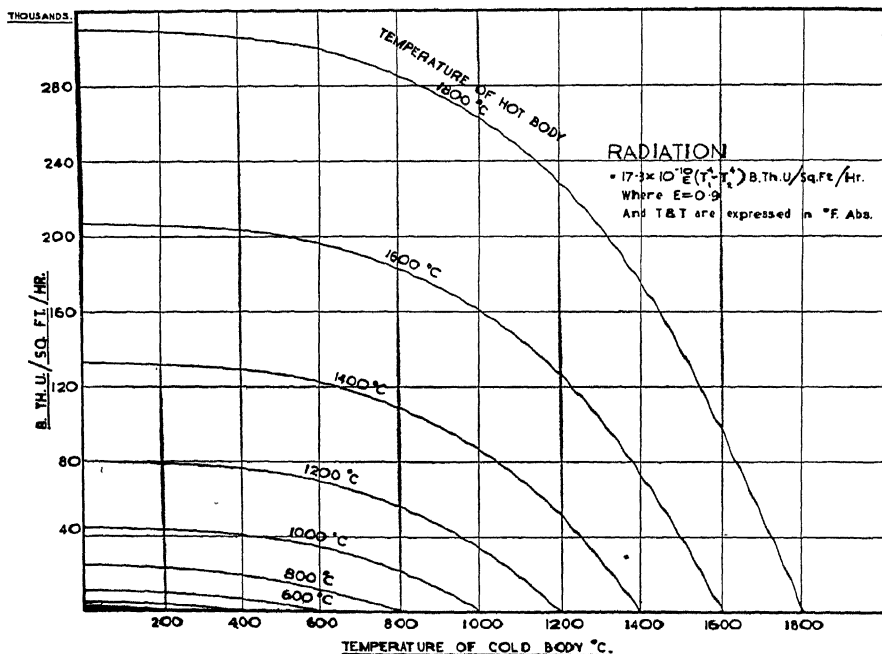


FIG. 42. Heat transfer by radiation from hot surfaces.

at 350° F., the value of E being 0.56. It can then be calculated that the heat transfer is as follows :—

Temperature of hot surface	..	{ °C. 800	1,000	1,200	1,500
		{ °F. 1,470	1,830	2,190	2,735
Heat transferred—H, B.Th.U./sq. ft./hr.	..	13,000	24,400	48,100	101,000
Relative values of H	..	1	1.88	3.7	7.8

It is thus evident that an increase in the hot surface temperature of less than double, i.e. 800°–1,500 C°, increases the rate of radiation nearly eightfold. Similarly an increase of 50 per cent. from 1,000° to 1,500° C. increases the rate fourfold.

These factors are further illustrated in Fig. 42.

The high rate of radiation from luminous flames is due to the radiating surface of the carbon in these flames, and clearly the higher the temperature to which the carbon is heated, the greater will be the radiating effect.

## RADIATION FROM A COKE FIRE

An example of the heat transfer from the surface of an incandescent fuel bed of coke, which is frequently used for heating vessels such as evaporating pans is given in Table 41. From practical observations it has been established that temperatures of the surface of the fuel bed corresponding to rates of combustion of coke of 20 up to 106 lb. per square foot per hour are those given in the second line of the table, namely temperatures ranging from 1,232° to 1,509° C. The temperatures of the heat-absorbing surfaces range between 200° and 400° C. in such practice. The heat emission, according to the fourth power law, has been calculated for the two temperatures concerned by the use of equation 27 and an emissivity coefficient of 0.9. The total potential heat in the coke corresponding to the rates of combustion applying are also shown, together with the radiant heat emitted reckoned as a percentage of this total value. The calorific value of the coke has been taken as 12,500 B.Th.U. per lb. It is to be noted that at the lower rates of combustion the direct radiant heat is approximately 33 per cent. of the total available heat, but that at the higher rates of combustion the proportion of radiant heat approaches a much lower value, e.g. 12 per cent. This is due to the fact that at the higher rates of combustion the mass of the gas emerging from the fuel bed is increased, and carries with it a higher proportion of the total available heat as either potential or sensible heat in the hot gases.

TABLE 41. RADIATION FROM A COKE FIRE

Rate of combustion—lb./sq. ft./hr.	20	51	71	106
Temperature of coke bed—° C. .. ..	1,232	1,383	1,442	1,509
Heat emission by radiation to surface at 200° C. ..	83.3	122.3	140.9	164.3
at 400° C. ..	80.5	119.5	138.2	161.5
(Thousands B.Th.U./ sq. ft./hr.)				
Rate of total heat release from coke .. ..	250	637.5	887.5	1,325
(Thousands B.Th.U./sq. ft./hr.)				
Per cent. of heat radiated to total heat developed				
Cold surface at 200° C. .. ..	33.3	19.2	15.9	12.4
at 400° C. .. ..	32.2	18.8	15.6	12.2

### GAS RADIATION

#### RADIATION OF CARBON DIOXIDE AND WATER VAPOUR IN FURNACE GASES

A further type of heat transference, which was ignored until relatively recent years is the direct radiation from hot furnace gases. It was found that in applying the ordinary methods of computation of heat exchange by convection to fire-tube boilers or to certain types of industrial furnaces, the surface conductances were of the order of magnitude of 10 (B.Th.U./sq.ft./hr./°F.). In practice surface conductances of a much higher order were observed. This led Schack to look for the cause of the discrepancy in the infra-red radiation of furnace gases, and the radiative capacity of carbon dioxide and water vapour always present were shown to account for the high practical values of the conductances.

Hot gases radiate heat even when no combustion is taking place. Good radiating gases include carbon dioxide, steam and hydrocarbons. Carbon monoxide has some radiating power, but is a relatively poor radiator. From a practical point of view hydrogen, oxygen and nitrogen are non-radiators. The radiation effect of a non-luminous flame is accordingly governed by its carbon dioxide and water vapour content.

The transfer of heat from the gas to the wall is an exchange process, since heat is also returned from the wall to the gas. On account of the thickness of the layer of gas some of the heat radiated by molecules of carbon dioxide and water are absorbed by other molecules before it can reach the wall. The course of the heat exchange in a combustion chamber containing the non-luminous products of combustion is as follows:—

(1) The gas radiation to the walls of the chamber depends upon the temperature of the gaseous atmosphere, and on the nature and amount of radiating gas between the walls, that is to say, on the concentration of the carbon dioxide and steam present, on the thickness of the gas layer and the pressure and temperature of the gas.

(2) Of the heat radiated by the gases only the fraction  $E$ , the emissivity of the material of the wall at the temperature ruling, is retained by the wall, the remainder being reflected back.

(3) At the same time there is heat exchange between the hot walls, radiating according to the normal Stefan-Boltzmann law, if temperature differences exist. Only a relatively low proportion of this heat is absorbed by the gases. If the wall is hotter than the gas the gas becomes heated; if the gas is hotter than the wall, the wall becomes heated by gas radiation.

The most convenient means of evaluating gas radiation is that due to H. C. Hottel.\*

The method involves the evaluation of the gas emissivity coefficients and applying them to a fourth power law in a manner which corrects for the influence of the gas composition in regard to the content of carbon dioxide and water vapour, and for the thickness of the gas layer.

The nomenclature used in describing the method, which involves the use of the diagrams, Figs. 43-46, is as follows:—

- $q$  = rate of heat transfer due to gas radiation, B.Th.U. per hour.
- $A$  = area of surface exchanging radiation with the gas, square feet.
- $T_g$  = absolute temperature of the gas ( $460 + t_g$  °F.).
- $T_s$  = " " " " surface ( $460 + t_s$  °F.).
- $E_g$  = emissivity of gas, dimensionless.
- $\alpha_g$  = absorptivity of gas, " "

\* "Heat Transmission," W. H. Adams (McGraw-Hill, 1942), p. 64.

- $p_c$  = partial pressure of carbon dioxide, atmospheres.  
 $p_w$  = " " " " water vapour, atmospheres.  
 $L$  = beam length of gas radiation, feet.  
 $E_s$  = emissivity of heat exchanging surface.  
 $E'_s$  = effective emissivity =  $\frac{E_s + 1}{2}$

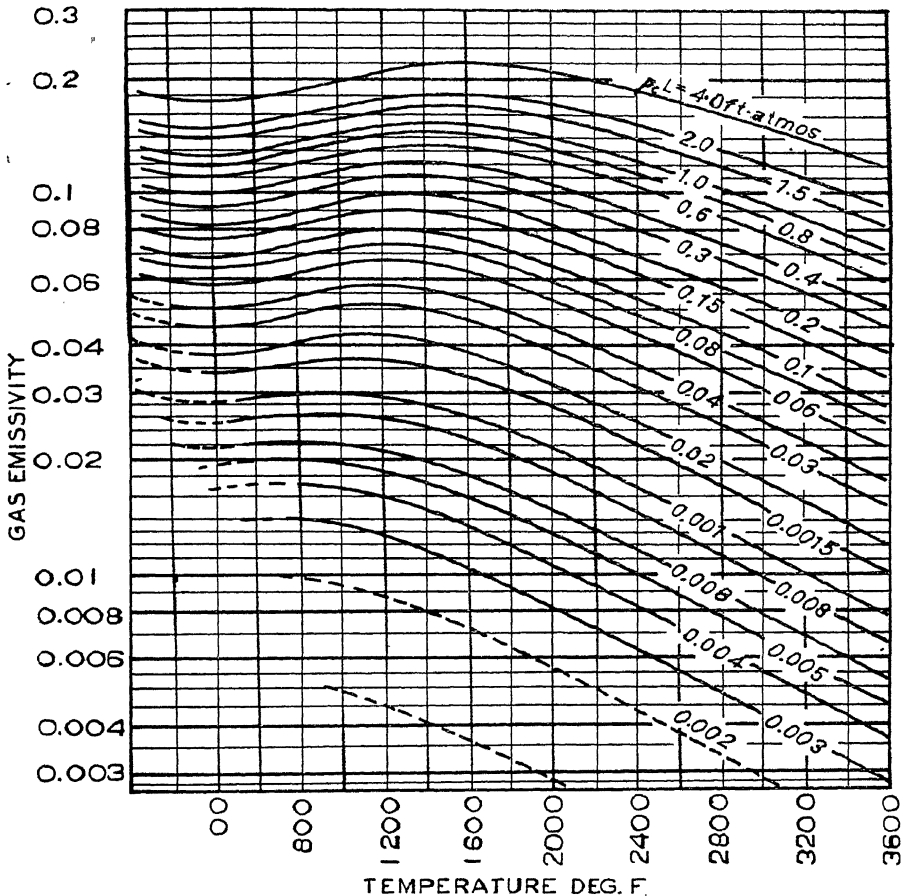


FIG. 43. Emissivity of  $\text{CO}_2$  for various values of  $pL$ .

(Reproduced by permission of the publishers (McGraw-Hill) from "Heat Transmission," by W. H. McAdams, 2nd edition.)

The complete calculation is complicated, but although it can be shortened in certain circumstances it will be given in full here step by step.

A relation, originally developed by Schack and amended by Hottel, exists between gas emissivity, temperature, and the product of the partial pressure of the gas and the thickness of the gas layer that is emitting the radiation. This relationship is the basis of Figs. 43 and 44.



From these figures, subject to certain corrections for which Figs. 45 and 46 are used, it is possible to compute the coefficient of emissivity of the gas  $E_g$ .

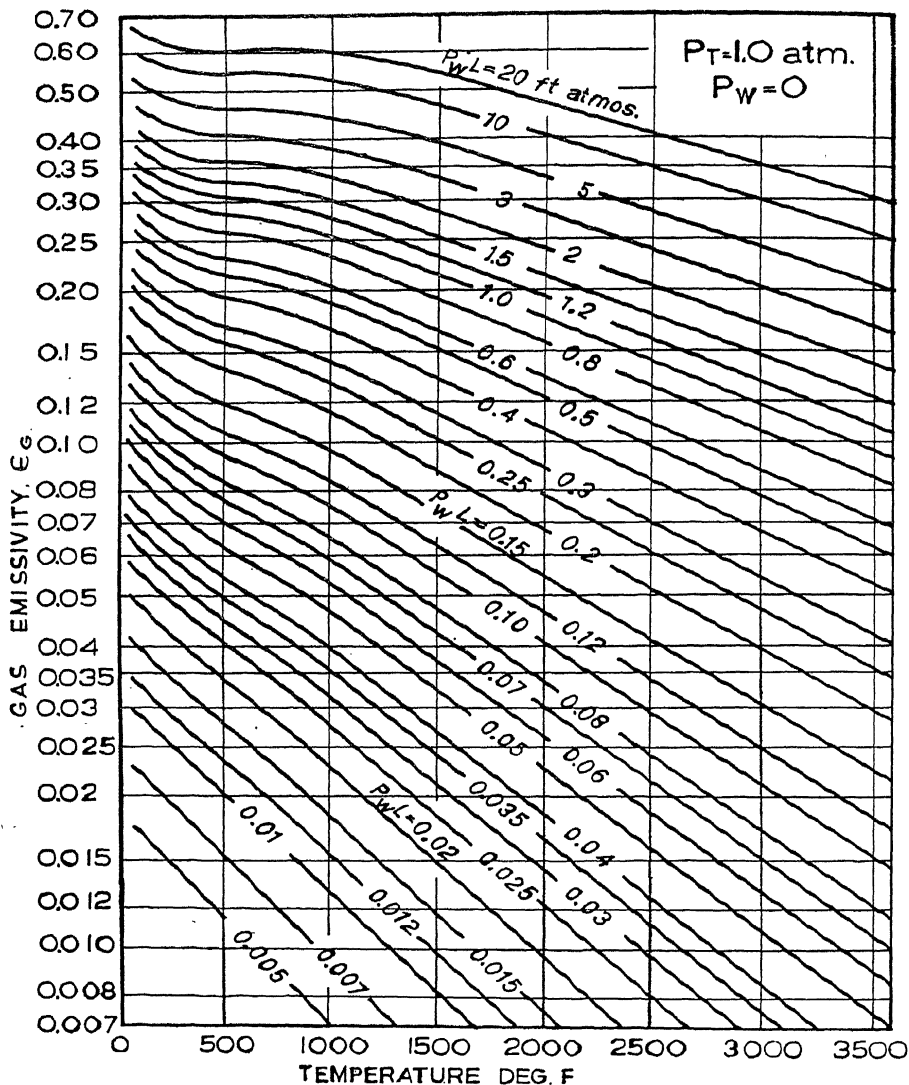


FIG. 44. Emissivity of water vapour for various values of  $p_w L$ .

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As will be explained later, the effective emissivity of the surfaces must also be taken into account, this being  $E'_s$ .

The gases absorb radiation emitted from the walls, and the value of the coefficient of absorption of the gas for this radiation,  $\alpha_g$ , must also be determined.

These coefficients are then used in equations 27 or 27a as follows:—

$$H = 0.173 \cdot E_s \left[ E_g \left( \frac{T_g}{100} \right)^4 - \alpha_g \left( \frac{T_s}{100} \right)^4 \right] \text{ B.Th.U./sq.ft./hr.} \quad (39)$$

The steps by which this calculation is made will now be given in order.

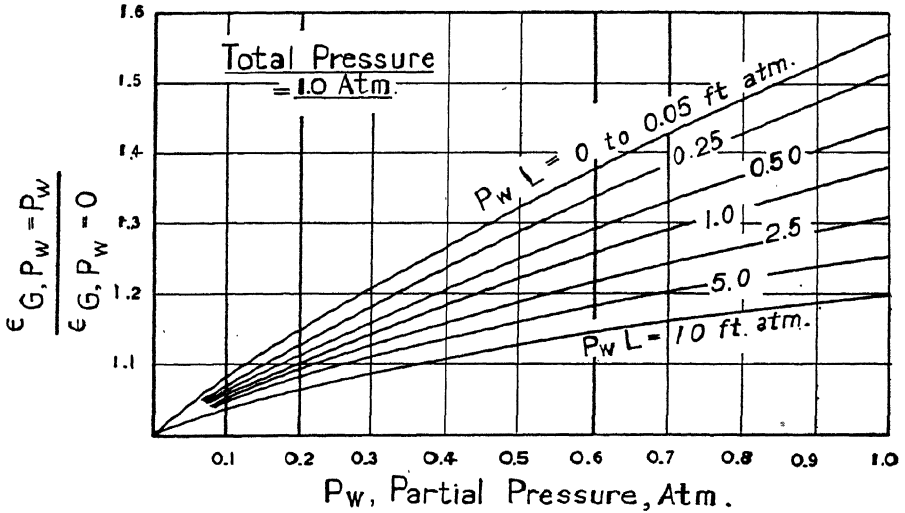


FIG. 45. Chart for evaluating gas radiation factor,  $C_1$ .

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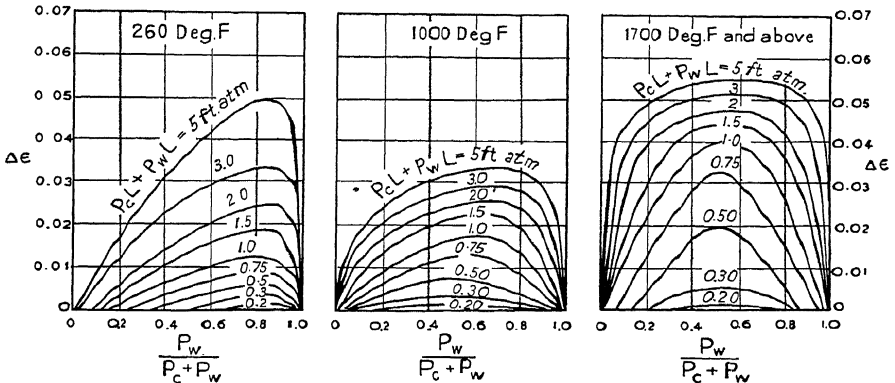


FIG. 46. Chart for evaluating the absorption of gas radiation by the gas itself.

(Reproduced by permission of the publishers (McGraw-Hill) from "Heat Transmission," by W. H. McAdams, 2nd edition.)

- (1) The partial pressure of the radiating gases  $\text{CO}_2$  and  $\text{H}_2\text{O}$  in atmospheres is determined.
- (2) The effective thickness of the gaseous layer is determined. In Figs. 43 and 44, the same thickness of gas in all directions is assumed. This is not generally the case and the effective thickness,  $L$ , is found from the general expression

$$L = \frac{4 \times \text{vol. of gas space}}{\text{area of bounding walls}}$$

Table 42 enables this value to be determined for several industrially important shapes.

TABLE 42. EQUIVALENT THICKNESS  $L$  FOR NON-LUMINOUS GAS RADIATION FROM LAYERS OF DIFFERENT SHAPES

Shape	Characteristic dimension, $X$	Factor by which $X$ is to be multiplied to give equivalent $L$ for hemispherical radiation	
		Calculated by various workers *	Calculated from $4 \times 0.85 \times \text{volume} \div \text{area}$
Sphere .. .. .	Diameter	0.60	0.57
Cube .. .. .	Side	0.60	0.57
Infinite cylinder, radiating to walls ..	Diameter	0.90	0.85
Ditto, radiating to centre of base ..	Diameter	0.90	0.85
Cylinder, height = diameter, radiating to whole surface .. .. .	Diameter	0.60	0.57
Ditto, radiating to centre of base ..	Diameter	0.77	0.57
Space between infinite parallel planes ..	{ Distance } apart	1.80	1.70
Space outside infinite bank of tubes with centres on equilateral triangles, tube diameter = clearance .. .. .	Clearance	2.80	2.89
Ditto, but tube diameter = one-half clearance .. .. .	Clearance	3.80	3.78
Ditto, with tube centres on squares, and tube diameter = clearance .. .. .	Clearance	3.50	3.49
Rectangular parallelepiped, $1 \times 2 \times 6$ , radiating to :—	Shortest edge	{ 1.06 1.06 1.06 1.06	{ 1.01 1.05 1.01 1.02
2 $\times$ 6 face .. .. .			
1 $\times$ 6 face .. .. .			
1 $\times$ 2 face .. .. .			
All faces .. .. .			
Infinite cylinder of semicircular cross-section, radiating to centre of flat side	Diameter	0.63	0.52

\* Eckert, 1937 ; Hottel, 1927 ; Hottel and Egbert, 1941 ; Jakob and Erk, 1933 ; Lander, Fishenden, and Saunders, 1936 ; Nusselt, 1926.

- (3) The values of  $p_c L$  and  $p_w L$  are then determined.
- (4) The coefficient of emissivity due to  $\text{CO}_2$ , i.e.  $E_c$ , is then read off from Fig. 43 for the ascertained value of  $p_c L$  at the required gas temperature.
- (5) In the same way the value of  $E_w$  is read off from Fig. 44. This value, however, is subject to a correction because the emissivity of water vapour,  $E_w$ , does not depend only on the temperature and on  $p_w L$ , but also on the concentration of water vapour in the gases. Fig. 44 is drawn for the ideal condition of  $p_w = 0$ , and allowance for the real value of  $p_w$  must be made by using Fig. 45, the value of  $E_w$  read from Fig. 44 being multiplied by a factor  $C_1$  read from Fig. 45.
- (6) The combined values of  $E_c$  and  $E_w$  represent the radiating power of the gas, but since substances that radiate also absorb radiation, a further correction must be made for the radiation absorbed by the gas. This correction,  $\Delta E$ , is read from Fig. 46.

Then,

$$E_g = E_c + E_w - \Delta E.$$

- (7) By exactly similar calculations, the value of the absorptivity coefficient,

$$\alpha_g = \alpha_o + \alpha_w - \Delta\alpha,$$

is determined, using the temperature of the surface, not that of the gas, when reading the coefficients from Figs. 43 and 44.

This method of determining  $\alpha_g$  is correct if the gas is appreciably hotter than the surface and covers most furnace conditions. If a more accurate method is needed, the values used for reading from Figs. 43 and 44 are not  $p_c L$  and  $p_w L$ , but these values multiplied by the ratio: (temperature of surface/temperature of gas)

i.e.  $p_c L \times (T_s/T_g)$  and  $p_w L \times (T_s/T_g)$

The values thus obtained are then multiplied by  $(T_g/T_s)^{0.65}$ .

- (8) If the bounding surfaces differ substantially from black body conditions, the factor  $E'_s$ , the emissivity of the surface must also be introduced as in equation 29. Since some of the gas radiation initially reflected from the surface would have further opportunity for absorption at a bounding surface, because much of the reflected radiation passes unabsorbed through the gas, the effective value of  $E'_s$  lies between the value for the emissivity of the surface and that of a black body. For most industrial applications, therefore,

- (9) Equation 39 is then applied.

A comprehensive account of this subject will be found in "Heat Transmission" by McAdams (McGraw-Hill, 1942). Table 42 is taken from the paper on heat transmission by Professor Lander (*loc. cit.*).

The method of calculation will be made clear by an example taken in the same steps as the description above.

It is required to ascertain the gas radiation to the crown of a furnace chamber under the following conditions:—

Dimensions of furnace 12 feet long  $\times$  4 feet wide  $\times$  2 feet mean height

Composition of gas:  $\text{CO}_2$  13.2 per cent.

$\text{H}_2\text{O}$  12.1 " "

$\text{O}_2$  and  $\text{N}_2$  74.7 " "

Mean temperature of gases : 2,000° F. (1,093° C.)

Mean temperature of furnace crown : 1,500° F. (816° C.)

- (1) Partial pressures :  $\text{CO}_2$  0.132 atmos.

$\text{H}_2\text{O}$  0.121 " "

- (2) From Table 42, the shortest edge is 2 feet, and the factor by which it must be multiplied is 1.06, thus  $L = 2.12$ .

- (3)  $p_c L = 0.132 \times 2.12 = 0.28$   
 $p_w L = 0.121 \times 2.12 = 0.256$

- (4)  $E_c$  (from Fig. 43) = 0.091.

- (5)  $E_w$  (from Fig. 44) = 0.084.

The factor,  $C_1$ , from Fig. 45 is 1.08.

$$E_w = 0.084 \times 1.08 = 0.091.$$

- (6) Correction for gaseous absorption (Fig. 46)  $\Delta\alpha = 0.022 = \Delta E$ .

Then,

$$E_g = 0.091 + 0.091 - 0.022 = 0.160.$$

(7) Determination of  $\alpha_G$ .

$$\alpha_e = p_e L \times (T_s''/T_G) = 0.28 \times 1,960/2,460 = 0.223$$

Emissivity at 1,500° F. read from Fig. 43 = 0.10

$$\alpha_e = 0.10 (2,460/1,960)^{0.65} = 0.116$$

$$\alpha_w = p_w L \times (T_s/T_G) = 0.256 \times 1,960/2,460 = 0.204$$

Emissivity at 1,500° F. read from Fig. 44 = 0.090

$$\alpha_w = 0.09 (2,460/1,960)^{0.65} \quad C_1 = 0.104 \times 1.08 \text{ (from 5)} \\ = 0.112$$

From Fig. 46,  $\Delta\alpha = \Delta E$  at 1,500° F.

$$= 0.02$$

Whence,

$$\alpha_G = 0.116 + 0.112 - 0.020 \\ = 0.208$$

(8)  $E_S'$  for a furnace interior may be taken as 0.9

$$H = 0.173 \times 0.9 [0.160 (2,460/100)^4 - 0.208 (1,960/100)^4]$$

$$= 4,343 \text{ B.Th.U./sq.ft./hr.}$$

Hence for these conditions, the coefficient of gas radiation is  $4,343/500 = 8.686 \text{ B.Th.U./sq.ft./hr./}^\circ\text{F.}$ , exclusive of the effect of radiation transmitted to the roof by refractory surfaces.

## RADIATION FROM LUMINOUS FLAMES

Non-luminous flames transmit energy chiefly by the infra-red radiation of carbon dioxide and water vapour. Luminous flames are produced by restricting the air supply so that the hydrocarbons present in the gas in the absence of sufficient oxygen for combustion are cracked to produce solid particles of carbon. The combustion of a luminous flame is consequently more protracted than that of a similar non-luminous flame; that is to say the flame arising from a similar flux of combustible matter is longer. The small particles of carbon absorb heat from the surrounding atmosphere of hot gas and radiate energy to the enclosing surface.

The effect of luminous radiation from flame presents a more difficult subject in which to relate any quantitative analysis of the heat emission to the effect produced in the furnace. Lindmark and Edenholm have approached the subject by an application of the fourth power radiation law to the incandescent particles of carbon in flames produced by oil and by pulverised coal. The inflation of the coal particles during combustion, producing cenospheres having a volume as great as eightfold that of the parent particles, has been associated with a favourable influence on flame radiation from the pulverised fuel.

Lent and Thomas, investigating the influence of benzole added to blast furnace gas under conditions so regulated as not to raise the flame temperature, found that radiation from the flame was four times that emitted from the non-luminous flame, and practically the same as from a black body. The value of luminosity has been appreciated to the extent of injecting hydrocarbons into lean gases. As the flame thickness, or the density of suspended carbon, decreases the emitted radiation decreases, and therefore the apparent radiation temperature falls and differs to a greater extent from the true temperature.

Schack described an instance of a rolling mill furnace heated by coke-oven gas in which, although the rendering of the flame luminous produced a drop of temperature of 180° F., the surface temperature of the crown increased by 145° F. At first in such a furnace the heat transfer to the relatively cold walls is high and the temperature of the waste gas low. Ultimately, however, the temperature of the indirect heating surfaces increases, and the heat transferred to these surfaces decreases as a result of the decreased difference of temperature between the combustion gases and the heat surfaces. Then the temperature

of the waste gas correspondingly increases. These observations gave rise to the suggestion that improvement of practice might arise from periodic carburetting of the flame. This is, of course, what happens in hand fired coal-burning furnaces.

The analysis of total heat transfer in furnaces operating at reheating temperatures ( $1,150^{\circ}$ – $1,300^{\circ}$  C.) has shown the following distribution between the various modes of heat transmission.

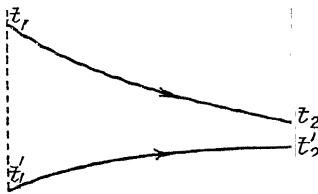
		Percentage of total heat transfer	
Direct gas radiation	..	..	54
Furnace wall and crown radiation	..	..	34
Convection	.. ..	..	12

Heat transfer by convection has only a subordinate influence in such furnaces.

### HEAT EXCHANGERS

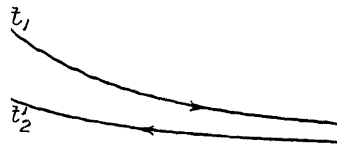
Heat exchangers are usually classified into two groups:—

- (1) those in which heat is transferred from one flowing material to another through a solid wall; and



HEATING SURFACE

FIG. 47. Temperatures in a parallel-flow heat exchanger.



HEATING SURFACE

FIG. 48. Temperatures in a counter-flow heat exchanger.

- (2) those in which heat is stored for some time and the heating gases and heated gases alternate in direction through the heat storage appliance.

To the first class belong recuperators, condensers, air heaters and economisers; to the second regenerators, usually applied to the preheating of gas and air in high temperature melting furnaces and in blast furnace stoves.

(1) *Heat Exchangers Without Heat Storage.* Heat exchangers may be operated either in parallel or counter-flow, and the temperature conditions along the heating surface are indicated in Figs. 47 and 48.

One material may flow at right angles to the second and the character of the flow is then referred to as cross-flow.

Under conditions of constant heat transmission, that is with no marked fluctuations of temperature, the simplest case by way of introduction only to a complicated study is that of two gases flowing through a heat exchanger with the temperature conditions as specified graphically in Fig. 47.

- If
- $q$  = heat transfer, B.Th.U. per hour.
  - $H$  = transmittance, B.Th.U./sq.ft./hr./ $^{\circ}$ F.
  - $A$  = total heating surface, square feet.
  - $t_g - t_a$  = mean temperature difference, between the flowing gases,  $^{\circ}$ F
  - $w$  = mass flow of the hotter gas, lb. per hour.
  - $c$  = specific heat of the hotter gas, B.Th.U./lb./ $^{\circ}$ F.
  - $w'$  = mass flow of the colder gas, lb. per hour.
  - $c'$  = specific heat of the colder gas, B.Th.U./lb./ $^{\circ}$ F.

Then the heat transfer referred to the gas giving up heat is

$$q = wc (t_1 - t_2) \quad (30)$$

and to the gas receiving heat

$$q = w'c' (t'_2 - t'_1) \quad (31)$$

The heat transfer is also governed by the transmittance  $H$ , the heating surface and the mean temperature difference between the flowing gases. Accordingly

$$q = H.A. (t_g - t_a) \quad (32)$$

Since the heat given up must be equal to the heat absorbed the right-hand sides of the equations 30, 31 and 32 must be equal to each other and

$$HA (t_g - t_a) = wc (t_1 - t_2) = w'c' (t'_2 - t'_1) \quad (33)$$

from which the heating surface  $A$  is given by either

$$A = \frac{wc (t_1 - t_2)}{H (t_g - t_a)} \quad \text{or} \quad \frac{w'c' (t'_2 - t'_1)}{H (t_g - t_a)} \quad (34)$$

The mean temperature difference  $t_g - t_a$  is for parallel flow

$$= \frac{(t_1 - t'_1) - (t_2 - t'_2)}{2.303 \log \frac{t_1 - t'_1}{t_2 - t'_2}} \quad (35)$$

and for counter-flow

$$= \frac{(t_1 - t'_2) - (t_2 - t'_1)}{2.303 \log \frac{t_1 - t'_2}{t_2 - t'_1}} \quad (36)$$

Arithmetical means for  $t_g - t_a$  are applicable within certain limits of accuracy.

The following example illustrates briefly an application. In a counter-flow air recuperator of 4-inch gas channels and 3-inch air channels separated by refractory tiles  $1\frac{1}{2}$  inches thick, waste combustion gases containing 13 per cent.  $\text{CO}_2$ , 6 per cent.  $\text{H}_2\text{O}$  and 5 per cent  $\text{O}_2$  are used to preheat air. The volumes of waste gases and air flowing are respectively 35,000 and 32,000 cubic feet per hour measured at  $60^\circ \text{F}$ . The waste gas enters the recuperator at  $1,470^\circ \text{F}$ . ( $799^\circ \text{C}$ .) and air entering at  $70^\circ \text{F}$ . is required to be preheated to  $1,000^\circ \text{F}$ . What is the temperature of the waste gases at the exit from the appliance and what is the area of heating surface required?

The problem involves a knowledge of the heat capacity of the gases and air in the ranges of temperature involved, and of the overall coefficient of heat transmittance  $H$ , which normally has to be calculated from the surface conductances by convection and gas radiation on the waste gas side, the conductance through the refractory tile and the surface conductance on the air side. For the sake of simplicity in explanation, the value of  $H$  is taken to be 1 B.Th.U./sq.ft./hr./ $^\circ \text{F}$ ., and the volumetric specific heats respectively 0.0215 and 0.0197 B.Th.U./cu.ft./hr. for waste gas and air.

Accordingly working in terms of volumes of gas instead of weights which were used in equations 32-34

$$q = 0.0215 \times 35,000 (1,470 - t_2) = 0.0197 \times 32,000 (1,000 - 70) \\ = 586,270 \text{ B.Th.U. per hour.}$$

$$\text{from which } t_2 = 691^\circ \text{F.}$$

From the complete knowledge so gained of the temperature conditions  $t_g - t_a$  (equation 36) becomes  $539^\circ \text{F}$ . and

$$A = \frac{q}{H (t_g - t_a)} = \frac{586,270}{1 \times 539} = 1,087 \text{ square feet.}$$

(2) *Heat Exchangers with Heat Storage Regenerators.* A regenerator usually consists of a mass of chequer brickwork built in a separate chamber through which the waste gases from the furnace descend to heat the chequers. They then pass to a reversing valve on their way to the chimney. A regenerative system in reheating and melting furnaces and certain types of kiln is applied to furnaces having two sets of ports, one at each end of the furnace chamber, through which the direction of the gases is periodically reversed.

When air alone is preheated there are two regenerator chambers, one serving to take up the waste heat from the hot gases, and the other to preheat the air

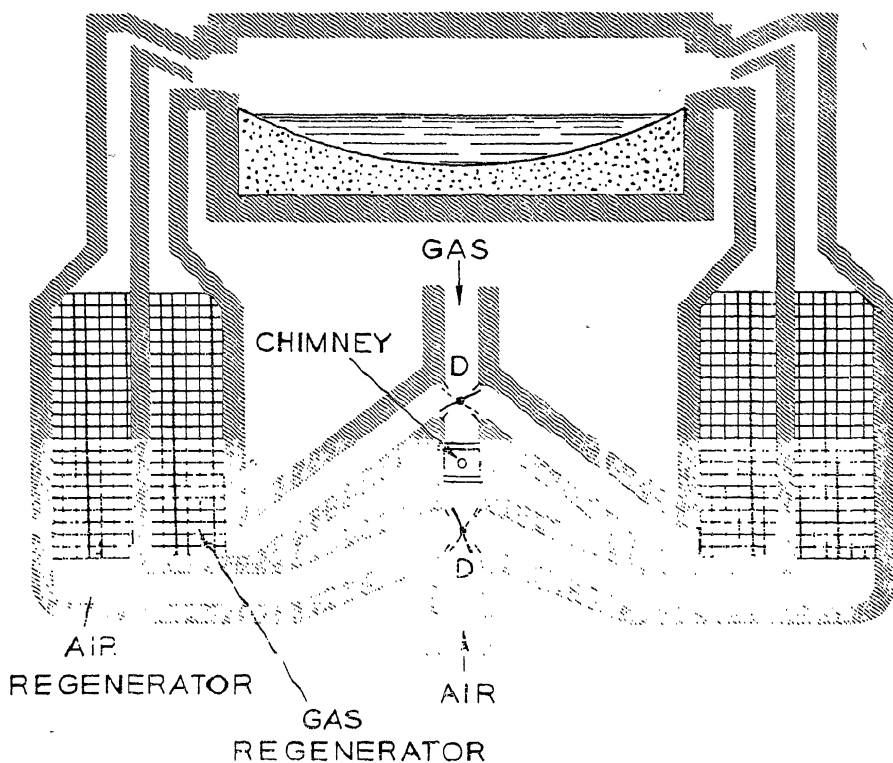


Fig. 49. Diagrammatic illustration of a regenerative furnace.

for combustion. Each regenerator has accordingly a heating cycle on waste gas and a cooling cycle on air. A blast furnace stove operates on the same principle except that gas is burnt in a combustion chamber built into the stove to provide the heat storage in the chequer brickwork during the heating cycle, the combustible gas being shut off during the cooling cycle when the blast is being preheated.

Fig. 49 depicts a regenerative system applied to an open-hearth steel furnace, or a glass furnace.

Here both gas and air are preheated. With the valves set as shown, air and gas flow up the left-hand regenerator where they are preheated by the hot chequer bricks to perhaps  $900^{\circ}$ – $1,000^{\circ}$  C. Each issues through its appropriate ports and combustion takes place in the furnace. The products of combustion



pass down through the right-hand regenerators, giving up their heat to the chequers, and then flow to the chimney. After a suitable period, the valves are changed to the dotted positions, when the flow of the gas and air is reversed. The cold gases now pass through the hot right-hand regenerators and the hot products of combustion leave the system through the cooled left-hand regenerators.

The problem of calculating the heat exchange is beyond the scope of this volume, since its solution involves the study of the fluctuating flow of heat. The subject has been adequately dealt with by Heiligenstaedt, Schack and Rummel, and applications to practice have been given by Barnes and Sarjant.

The implications of these brief references to the calculation of heat exchangers to the reader not concerned with the technique of design is that the laws of heat transmission have been developed and applied with sufficient certainty to most of the appliances used in industry so that a definite performance can be forecast within specific limits of accuracy. Optimum conditions of operation can be defined, which will give the most efficient performance.

If, however, the plant is not operated to satisfy these conditions, a waste of fuel must result. It is important, therefore, for the duty required from a specific installation to be clearly understood, and the load on the plant adjusted to meet these requirements. For example, losses cannot be avoided in banked boilers or in under-driven or over-driven furnaces.

Makers of plant will always give adequate information upon the optimum conditions of operation, and the limits of performance which give rise to excessive losses, and it should be the aim of the user to be aware of these conditions and make use of them in setting the conditions under which the plant is operated.

### HEAT LOSSES

During the course of this chapter attention has so far been directed mainly to the second stage of the process of heating, that of transmitting heat to the objects that are required to be heated. The third stage must now be considered, that of preventing the escape of heat in directions where it will serve no useful purpose.

Heat can escape through the sensible heat of the flue gases or by conduction into the outer air. The loss of heat in the flue gases has been considered in earlier chapters. The present discussion will be concentrated upon the loss of heat through furnace walls, through the surface of steam pipes, hot water tanks and in general in any manner in which heat can be dissipated from industrial plant by the processes under discussion in this chapter—convection, conduction and radiation.

If a part of an industrial plant has a temperature higher than the atmosphere, heat will be conducted through the casing or outer wall by the normal process, depending upon temperature difference between the hot and cold surfaces of the wall, and the thermal conductivity of the material.

This heat will raise the temperature of the outer sides of the wall or casing above that of the atmosphere and its surroundings. The outer surface will then lose heat by convection to the air and by radiation to the cooler objects around in the manner previously described.

It is an essential characteristic of this form of heat loss that the quantity of heat conducted through the wall or casing must equal exactly the quantity of heat radiated and convected from the outer surface. If the steady state has not been reached and this condition is not fulfilled, the temperature of the outer surface will change until the condition is fulfilled; at this point the steady state will be set up. Some account of the industrial significance of heat losses in the unsteady state will be found in Chapter XVIII.

When the steady state has been reached it is thus possible to calculate the surface temperature of the wall. An example will make the methods of calculation clear and in order to give an example of practical value attention will be directed to the insulation of a furnace.

### HEAT LOSSES FROM SURFACES

The total radiation and convection is obtained by a combination of equations 15 and 27.

$$H_c + H_r = C(t_1 - t_2)^{1.25} + 17.3 \times 10^{-10} \times E(T_1^4 - T_2^4), \text{ B.Th.U./sq.ft./hr.} \quad (37)$$

As stated above, the temperature of the hot surface is normally governed not only by the character of the heat loss from the external surface, but also by the heat flux through the mass forming the wall.

The calculated values of the combined radiation and convection loss from a furnace wall in a still atmosphere at 70° F. are shown in Table 43 (J. S. F. Gard, *J. Inst. Fuel*, X, 250).

TABLE 43

Surface temp. ° F.	Heat loss B.Th.U./sq. ft./hr.		
	Upwards	Vertical	Downwards
100	54	48	41
150	172	152	128
200	316	280	236
250	494	436	360
300	696	618	526
350	926	825	708
400	1,190	1,070	930
450	1,510	1,360	1,190
500	1,850	1,680	1,480
550	2,250	2,050	1,810
600	2,700	2,470	2,200
650	3,200	2,940	2,640
700	3,770	3,480	3,140
750	4,370	4,050	3,680

Table 43 is based on the coefficient of emissivity,  $E$ , for firebrick (Table 40), and would require to be recalculated if the material with which the wall was coated had a different coefficient of emissivity. Actually the governing factor in determining heat losses through furnace walls is the insulating character of the wall.

The effect of the coefficient of emissivity of the material is shown in Fig. 50. By using a bright unscratched metal surface (e.g. nickel plated) the value  $E$  can be reduced to about 0.05 (cf. Table 40), and the heat loss by radiation becomes reduced to very small dimensions. The heat loss by convection is unaffected and the total heat loss is the sum of the convection and radiation effects for any given temperature. This device is employed in hospitals and other places where it is possible to maintain the surface in something approaching its initial bright condition.

The convection loss for the same temperature conditions is greatly affected by the air currents prevailing. Table 43 and Fig. 50 refer to still air. A suggested approximate expression to take air movement into account is

$$\text{loss of heat by convection} = (1.2 + 0.19V)(t_s - t_a), \text{ B.Th.U./sq.ft./hr.} \quad (38)$$

where  $V$  = air velocity in feet per second

$t_s, t_a$  = temperatures of surface and air, °F.

Table 44 (A. C. Pallot, *J. Royal Soc. Arts*, XCI, 123) shows another (and slightly higher) estimate of the amount of heat transfer in still and moving air respectively from a vertical surface ( $E = 0.9$ ) maintained at various temperatures, the air being at  $70^{\circ}\text{F}$ .

**B.Th.U./FT<sup>2</sup>/HR.**

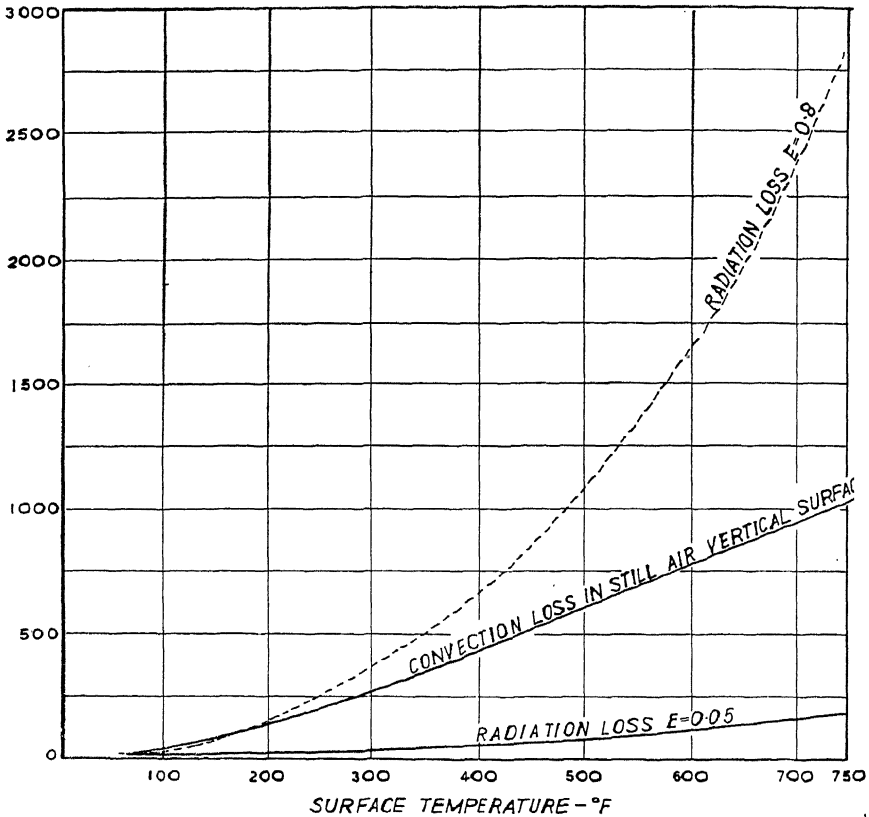


FIG. 50. Curves showing convection and radiation loss from surfaces in still air at  $70^{\circ}\text{F}$ .

TABLE 44. EFFECT OF AIR MOVEMENT ON HEAT LOSS FROM FURNACE.

Temp. of Surface ° F.	B.Th.U./sq.ft./hr.					
	Still air conditions			Air moving at 15 m.p.h.		
	Radiation	Convection	Total	Radiation	Convection	Total
100	30	23	53	30	162	192
200	169	140	309	169	700	869
300	390	287	677	390	1,240	1,630
400	715	450	1,165	715	1,780	2,495

The importance of properly housing furnace structures, hot water tanks, etc., and of ensuring that their outer surfaces are not subjected to draughts will be seen from this example.

### HEAT LOSS THROUGH A FURNACE WALL

The temperature of the interior surface of a furnace wall should be known, since it can be measured ; it should also presumably be reasonably constant, since it is a temperature at which the furnace is intended to be operated. To indicate the method of calculating the effect of insulation, a furnace is selected, the internal temperature of which is  $2,000^{\circ}\text{F.}$ , and having walls which are composed of  $13\frac{1}{2}$  inches of firebrick having a thermal conductivity,  $k$ , of 10 B.Th.U./sq.ft./hr./ $^{\circ}\text{F.}/\text{in.}$

The first step is to make an estimate at the outside furnace temperature ; this estimate may be  $600^{\circ}\text{F.}$  From the application of the conductivity formula,

$$H = \frac{10 (2,000 - t_2)}{13\frac{1}{2}}$$

it is calculated that when  $t_2$  is taken as  $600^{\circ}\text{F.}$ ,  $H = 1,035$  B.Th.U./sq.ft./hr. From Table 43 (vertical wall), it will be seen that with a surface temperature of  $600^{\circ}\text{F.}$  the surface loss is 2,470 B.Th.U. Evidently the estimate at the value of  $t_2$  was much too high.

If  $t_2$  be now taken as  $400^{\circ}\text{F.}$ ,  $H = 1,185$ , by the same method of calculation as before ; but by interpolating in Table 43 the surface loss is seen to be 1,070 B.Th.U./sq.ft./hr. This estimate is evidently slightly too low.

Proceeding in this way, it can be ascertained by trial and error (or more exactly by graphical methods) that at a temperature of  $420^{\circ}\text{F.}$ ,  $H = 1,170$ , and this virtually coincides with the surface loss from a vertical wall as given in the table.

The surface temperature of the wall is therefore  $420^{\circ}\text{F.}$  and the loss of heat is 1,170 B.Th.U./sq.ft./hr.

If now a  $4\frac{1}{2}$ -inch covering of insulating brick ( $k = 1.2$ ) is built outside the  $13\frac{1}{2}$ -inch firebrick wall, the formula for a composite wall must be used :—

$$h_k = \frac{10}{\frac{13\frac{1}{2}}{10} + \frac{4\frac{1}{2}}{1.2}} \text{ B.Th.U./sq.ft./hr.}$$

Proceeding in the same manner as before, it can be calculated that by adding  $4\frac{1}{2}$  inches of insulating brick the external surface temperature is reduced to  $225^{\circ}\text{F.}$  and the heat loss to 360 B.Th.U./sq.ft./hr.—another striking commentary on the value of insulation.

Some calculations on these lines are summarised in Table 45.

TABLE 45.

Furnace wall		Furnace wall temperature		
Firebrick inches	Insulating brick inches	Internal $^{\circ}\text{F.}$	External $^{\circ}\text{F.}$	Heat loss B.Th.U./sq. ft./hr.
13½	nil	2,000	420	1,170
9	nil	2,000	500	1,680
13½	4½	2,000	225	360
9	4½	2,000	230	380
13½	9	2,000	170	205
9	9	2,000	172	210

These figures indicate (a) the relative efficiency of thickening furnace walls to reduce heat loss ; (b) the outstanding value of insulating bricks for this purpose ; (c) the fact that a moderate thickness of insulating material achieves most of the thermal saving.

In a similar manner the temperature at the interface between the normal and the insulating refractory layer can be calculated, for example, for a composite wall of 9 inches of firebrick and 9 inches of insulating brick, cited in Table 45.

$$\frac{10 (2,000 - t_2)}{9} = \frac{1.2 (t_2 - 172)}{9}$$

whence  $t_2 = 1,804^\circ \text{F.} (984^\circ \text{C.})$

A knowledge of the temperature of the interface is required in order to determine whether the maximum safe temperature is reached to which the insulating refractory can be subjected without breakdown. External insulation always increases the average temperature of the inner refractory lining and may in extreme conditions cause this to be raised above its safe working temperature.

Fig. 51 shows a method for the rapid estimation of temperature gradients and heat loss through furnace walls due to General Refractories Ltd. The principle is to express the ratio between wall thickness and conductivity as a quantity  $x$ . The hot face temperature is known as being either the temperature at which the furnace is to operate or as being measurable. The surface temperatures, interface temperatures and heat losses can then be read from the chart.

Two examples will make the method clear :—

- (1) A furnace wall (hot face temperature  $1,600^\circ \text{C.}$ ) consists of 18 inches thickness of firebrick, conductivity 10 B.Th.U./sq.ft./hr./ $^\circ \text{F.}$ /in. thickness. It is required to calculate the external surface temperature and the heat loss (cf. "Example 1," Fig. 51).

$$x = \frac{\text{thickness}}{\text{conductivity}} = \frac{T/AB}{k/AB} = \frac{18}{10} = 1.8$$

Reading from Fig. 51 at the junction of the horizontal line  $x = 1.8$  and the curve for  $1,600^\circ \text{C.}$ ,

cold face temperature =  $210^\circ \text{C.}$

heat loss = 1,250 B.Th.U./sq.ft./hr.

- (2) A furnace wall (cf. example 2, Fig. 51) consists of 18 inches firebrick (conductivity 10)  $4\frac{1}{2}$  inches insulation (conductivity 1.2) and  $4\frac{1}{2}$  inches building brick (conductivity 6) and with a hot face temperature of  $1,600^\circ \text{C.}$  It is required to calculate the cold face temperature, the heat loss, and the interfacial temperatures.

$$x = \frac{B}{k/AB} + \frac{T/BC}{k/BC} + \frac{T/CD}{k/CD}$$

$$= \frac{18}{10} + \frac{4.5}{1.2} + \frac{4.5}{6} = 6.3$$

Reading from Fig. 51,

cold face temperature =  $120^\circ \text{C.}$

heat loss = 520 B.Th.U./sq.ft./hr.

If  $A^\circ$  = temperature at face A

$B^\circ$  =           ,,           ,,           B, etc. (Fig. 51, example 2)

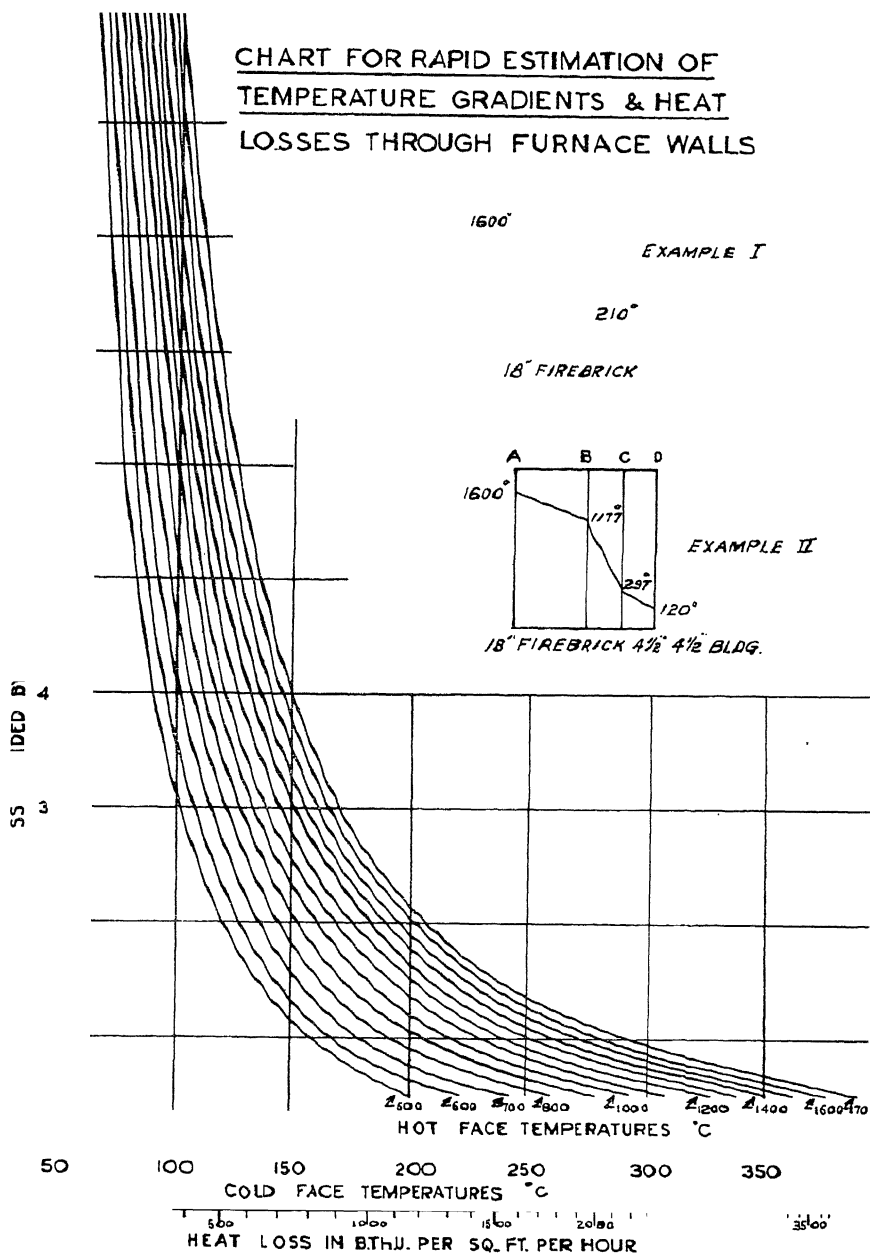


FIG. 51. Chart for rapid estimation of temperature gradients and heat loss through furnace walls.

$$\begin{aligned}
 & \left( \frac{A^\circ - D^\circ}{x} \times \frac{T/AB}{k/AB} \right) \\
 &= 1,600 - \left( 1,600 - \frac{120}{1} \times \frac{18}{10} \right) \\
 &= 1,177^\circ \text{C.} \\
 &= B^\circ - \left( \frac{A^\circ - D^\circ}{x} \times \frac{T/BC}{k/BC} \right) \\
 &= 1,177 - \left( \frac{1,600 - 120}{6.3} \times \frac{4.5}{1.2} \right) \\
 &= 297^\circ \text{C.}
 \end{aligned}$$

Insulating materials are discussed in Chapter XXII ; and furnace insulation technique in Chapter XVIII. These chapters include some account of the practice of hot face insulation.

### PRACTICAL ASPECTS OF HEAT TRANSMISSION IN BOILERS

#### TRANSMISSION BY CONDUCTION, CONVECTION AND RADIATION

The evaporation of the boiler is sometimes referred to in terms of the equivalent energy produced in the form of steam. Thus may arise the statement that a boiler rating is, say, 10 square feet of heating surface per boiler horsepower. This is equivalent to a rate of heat transmission of

$$\frac{34.5 \times 970.3}{10} = 3,350 \text{ B.Th.U./sq.ft./hr.}$$

the value of 34.5 being the pounds of water evaporated "from and at 212° F." in the generation of one boiler horse-power, and 970.3 being the latent heat of steam at 212° F. If the thermal conductivity of iron is 408 B.Th.U./sq.ft./hr./°F./in. thickness at a temperature of 400° F. and the boiler plate is  $\frac{1}{8}$  inch thick the temperature gradient across the plate is only

$$\frac{3,350 \times \frac{1}{8}}{408} = 0.82^\circ \text{F.}$$

In experimental trials with a Heine boiler, a difference of temperature of 41.5° F. between the two surfaces of a tube  $\frac{1}{8}$  inch thick was observed. Accordingly the heat transmission through the tube must have been

$$\frac{408 \times 41.5}{\frac{1}{8}} = 135,450 \text{ B.Th.U./sq.ft./hr.}$$

$$= 140 \text{ lb./sq.ft./hr. of steam "from and at 212° F."}$$

When these rates are compared with those which are actually operated in boiler practice it will be realised what are the potential capacities of clean surfaces, and to what extent the soot and scale coatings interfere with the rate of heat transmission. The above rate of transfer was equivalent to conditions of a fuel bed and furnace temperature of 2,640° F. and a sooted surface temperature of 800° F.

The data given in Fig. 52, after the Cochrane Corporation, gives a picture of the temperature conditions in a boiler tube when the initial temperature of the gases is 2,500° F. and the boiler is working at 10 times a normal rate. From this diagram it will be seen that the process of getting the heat by radiation and convection through to the dry surface of the boiler tube is slow compared with that on the wet side. The course of the temperature conditions through the gas passages of a boiler is shown in Fig. 53, based on experimental observa-

## HEAT TRANSMISSION

tions on water tube boilers. Heat is transmitted from the hot fuel bed and adjoining refractory surfaces to boiler heating surfaces by direct radiation, from the products of combustion by gas radiation, and by convection. The conductance by convection varies between 1 and 12 B.Th.U./sq.ft./hr./°F, difference for the gases flowing according to conditions of flow. The pheno-

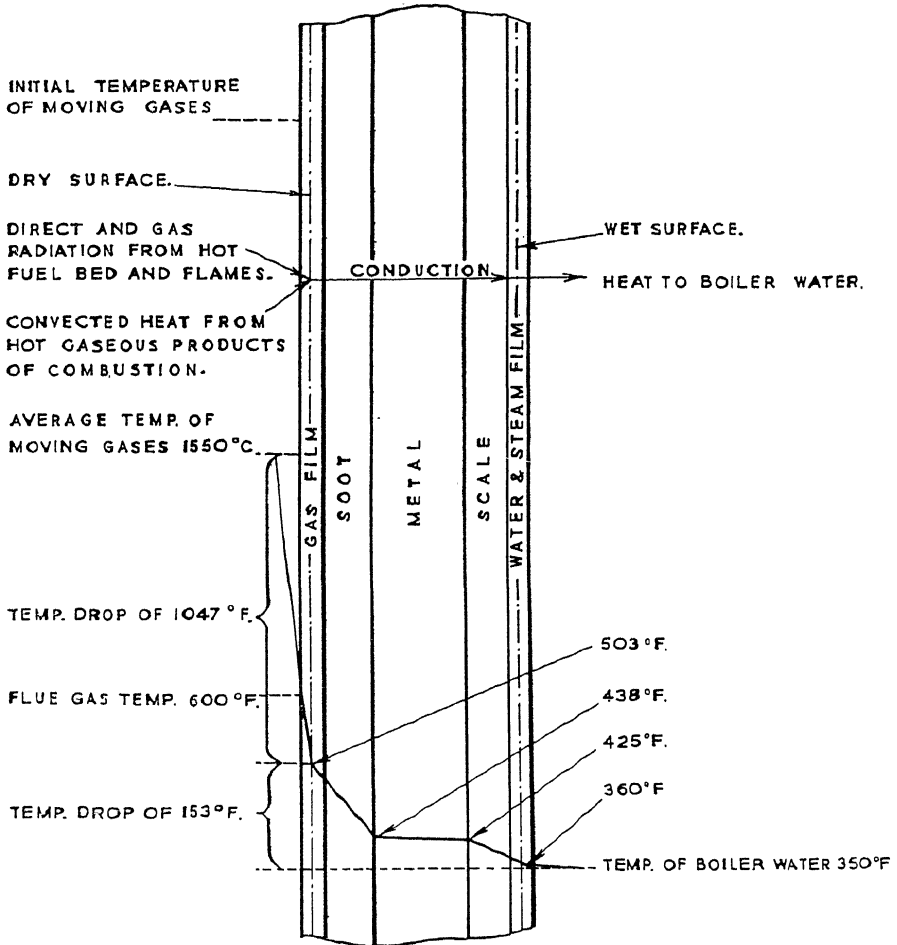


FIG. 52. Heat transmission from hot flue gases to boiler water

menon is illustrated most suitably by reference to waste heat boilers fitted with fire-tubes. These are usually equipped with high-power exhaust fans to give the high velocities associated with conditions of efficient heat transfer by convection.

The furnace of the locomotive boiler presents an adequate illustration of heat transfer by radiation. The hot surface is the fuel bed, and the cold surface the combined surface of the tube sheet, the crown sheet, the front sheet and the plates on both sides of the fuel bed. Increasing the furnace temperature from



2,000° F. to 3,000° F. will nearly quadruple the amount of heat imparted by radiation. A drop of temperature of 100° from 2,500° to 2,400° which readily arises from unskilful firing will reduce the amount of heat transferred by radiation by 12 per cent. This indicates the importance of maintaining the boiler furnace fires in a state of the brightest incandescence for efficient utilisation of the available heat of the fuel.

Bell showed experimentally how much heat is absorbed by a boiler by radiation by replacing a long tube in a water-tube boiler by a shorter one of the same

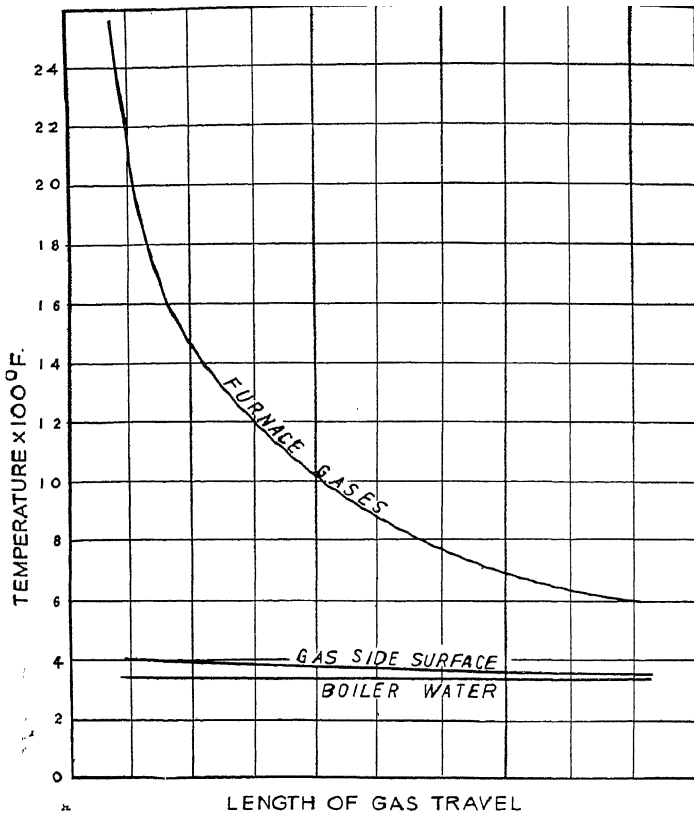


FIG. 53. Temperature drop through the gas passes of a water-tube boiler.

size extending through the front header, and just entering the first baffle. The heat absorption was determined from the temperature and mass of the water which was circulated independently through this separate tube. The heat absorbed was equivalent to a rate of evaporation of 75 lb. of steam per square foot per hour. Only a portion of the surface was heated by radiation, and it was estimated that this was raised to about 100 lb. per square foot per hour from the impingement of the gases (convection). As the gases were at about 2,500° F. the results were of an order which checked well with the fourth power radiation law. The water-cooled walls of large stoker-fired boilers make use of the direct radiation from the flame.

Broido, quoted by the Cochrane Corporation, has given the distribution of

heat transmitted from the gases flowing through a boiler at the rate of 6,000 lb. per hour per square foot of area, at two points where the gas temperatures were 1,400° F. and 800° F., respectively, steam pressure 250 lb. per square inch gauge, water and tube temperature about 406° F. The amount of heat transmitted by convection depends greatly on the velocity of the gas at the point under consideration. Broido's results are given in Table 46.

Where the gases fall in temperature in the gas circuit convection transfer may be predominant if the velocity is sufficiently great, but high rates of transmission require the use of high velocities.

TABLE 46

Gas temperature, °F.	1,400	800
Total heat transfer, B.Th.U./sq. ft./hr.	7,660	2,620
Total conductance coefficient—B.Th.U./sq. ft./hr./° F.	7.7	6.65
Gas radiation—CO <sub>2</sub> —B.Th.U./sq. ft./hr.	1,340	280
Gas radiation—H <sub>2</sub> O—	477	107
Total gas radiation	1,817	387
Heat transfer by convection (by difference) B.Th.U./sq. ft./hr.	5,843	2,233
Convection coefficient, B.Th.U./sq. ft./hr./° F.	5.83	5.66

One of the most complex problems of heat transmission is the evaluation of the performance of a combustion chamber of a furnace or boiler, in which heat is being transmitted simultaneously in all of the ways which have been discussed. Allowance must be made for the combined actions of direct radiation from the flame to the medium heated, the boiler plates or tubes and refractory surfaces, thence back through the flame, with partial absorption therein to the colder evaporation surfaces. Convection and external losses have also to be considered. The problem is beyond the scope of the present treatment, but the considerations involved are dealt with by H. C. Hottel.\* By the use of the same technique and a knowledge of the rate of fuel consumption and the evaporation of the boiler the distribution of the flow of heat can be traced throughout the boiler system.

## HEAT TRANSMISSION IN ECONOMISERS

An economiser is a water heater which is used to preheat the water entering the boiler by means of the sensible heat in the waste gases leaving the boiler. It is in effect a heat exchanger in which evaporation is not required, and a form of heating surface must be substituted more suited to the lower temperature conditions involving only heating of the water. The mode of heat transmission is that of a counter-flow heat exchanger in which a balance has to be struck between the rise in temperature of the water and the fall in temperature of the boiler outlet gases.

In an economiser of conventional design the temperature of the outlet gases is so adjusted by the allowance of heat exchanging surface as to prevent too close an approach to steam formation. In recent years steaming types of economisers have come into use and this limitation then no longer holds.

It has been shown in the discussion of heat transfer by convection, which is the predominant means of heat transfer from gases at low temperatures, that increase of gas velocity is necessary to raise rates of heat transfer. In the older installations where chimney draught was limited velocities had to be kept low in order to limit loss of draught due to the resistance of the tubes, and a large heating surface or a large number of tubes was required for a given duty. The efficiency of the heat transfer through the surface of the tubes was preserved by the fact that they were automatically scraped clean of soot.

\* W. A. McAdams, "Heat Transmission." McGraw-Hill (1942), Chapter 3.

The possible effect of the better heat transfer due to increased velocity naturally comes into question. This can be effected by modifying the design as, for example, in the vertical type of apparatus, and by the use of baffles. A heat transfer coefficient of 3 B.Th.U./sq. ft./hr./°F. thus becomes raised to values of the order of 5.5 to 7.7. Higher velocities tend to keep the tubes cleaner, and the limit in the development of velocity of the gases depends upon the available draught and the additional power to be provided where forced draught is employed.

### DRAUGHT LOSS IN RELATION TO ECONOMISER EFFICIENCY

The relation has been shown by G. E. Tansley\* in terms of temperature rise to be as follows :—

If  $W$  = weight of gases, lb. per hour  
 $w$  = evaporation, lb. per hour  
 $t$  = temperature of gases at fan, °F.

Then, volume of the gases weighing 0.076 lb./cu. ft. at 60° F.

$$\begin{aligned} &= \frac{1}{0.076} \times \frac{t + 460}{520} \\ &= 0.0252 (t + 460) \text{ cu. ft./lb.} \\ &= \frac{W}{60} \cdot 0.0252 (t + 460) \text{ cu. ft./minute.} \end{aligned}$$

Fan horse-power per 1 inch w.g. draught

$$= \frac{W \cdot 0.0252 (t + 460) \times 5.2}{60 \cdot 33,000}$$

For a motor efficiency of 90 per cent., and a fan efficiency of 50 per cent. the power input to the motor

$$= \frac{W \cdot 0.0252 (t + 460) \times 0.746 \times 5.2}{60 \times 33,000 \times 0.9 \times 0.5} \text{ kWh}$$

If the heat in the steam required per kilowatt-hour is taken as 20,000 B.Th.U., the heat equivalent of the motor input

$$= \frac{W \cdot 0.0252 (t + 460) \times 0.746 \times 5.2 \times 20,000}{60 \times 33,000 \times 0.9 \times 0.5} \text{ in.}$$

and the rise in the feed water temperature equivalent to 1 inch water gauge

$$\begin{aligned} R &= \frac{W (t + 460) \times 0.0252 \times 0.746 \times 5.2 \times 20,000}{w \times 60 \times 33,000 \times 0.9 \times 0.5} \\ &= \frac{W}{w} \cdot \frac{(t + 460)}{455} \end{aligned}$$

Actually the heat per kilowatt-hour may vary widely according to the efficiency of power generation. Further, it is not correct to say that a higher draught loss necessarily means a greater rate of heat transfer unless it is clearly understood that the form, type and arrangement of the heating surface remain constant. Further, the question of fan power involves economic considerations outside the heat transfer problem.

A related question may also be examined, viz., does the staggering of the tubes of an economiser confer any appreciable benefit? Here the generalisation that staggered tubes are better than non-staggered breaks down. The pitching of the tubes in an economiser, say  $4\frac{3}{8}$  inches diameter on 8-inch centres is so

\* *Trans. Inst. Mining Engineers*, LXXIX, Pt. 2, pp. 190–208.

wide and the gas velocities in general so low that the increase in both heat transfer and draught loss is quite negligible. The point has been subjected to lengthy examination, and the modern method for speeding up rate of gas travel in certain types of economisers is to arrange the tubes in groups with vertical baffles between, so that instead of being across them as in the normal arrangement, the flow of gases is approximately parallel to the length of the tube. Thus in an economiser of say 192 tubes, 8's wide, 9 feet long, the area for cross gas flow is 33·5 square feet. If this economiser is arranged in three groups of 64 tubes each, with vertical baffles between, the gas flow area becomes 23·18 square feet and with the same total gas quantity the gas speed is proportionately greater.

Both heat transfer rate and draught loss increase, but the latter is not a direct function of the former in this example, since part of the increased loss in draught is due to change of direction over and under the baffles. A typical set of performance figures will illustrate this point (Table 47).

TABLE 47. PERFORMANCE FIGURES FOR AN ECONOMISER

Economiser of 192—8's 9 ft.

Heating surface 2,234 sq. ft.

	Normal arrangement	In three groups with vertical baffles
Weight of water, lb./hour .. .. .	20,000	20,000
Weight of gases, lb./hour .. .. .	45,000	45,000
Gas flow area—sq. ft. . . . .	35·5	23·18
Gas velocity in lb./hour/sq. ft. of flow area ..	1·345	1·945
Rate of heat transfer in B.Th.U./sq. ft./hr./° F. ..	3·5	3·8
Inlet gas temperature—° F. . . . .	650	650
Outlet gas temperature—° F. . . . .	407	394
Inlet water temperature—° F. . . . .	120	120
Outlet water temperature—° F. . . . .	244	251
Total draught loss due to friction and change of direction, in. w.g. . . . .	0·07	0·24
Draught loss due to change of direction, in. w.g. ..	—	0·08

For this particular set of conditions, therefore, there was shown a gain in performance represented by 7° F. rise in water temperature, as against an extra 0·17 inch w.g.

How far this gain in performance is justified depends again upon circumstances involving a wide range of considerations, operational and economic. If, for example, induced draught enables an inferior and formerly unusable fuel to be used, some proportion of the fan charges should logically be added to the fuel cost, and not debited entirely to the economiser saving.

The wider aspects of these problems are dealt with in the later chapters on boilers and their auxiliaries.

## CHAPTER IX

### THE FLOW OF FLUIDS

The mechanics of fluid flow—Bernoulli's theorem—Streamline and turbulent flow—Viscosity of fluids—Equations of fluid flow—Enlargement and contraction losses—Effect of bends, etc.—Special formulae used for gas transmission—Principles of the measurement of fluid flow—The flow of gases in furnaces—Examples of fluid flow calculation.

**F**LUID flow is a highly important branch of fuel technology. Liquids, vapours (including steam) and gases have to be transported in pipes, an operation involving the expenditure of power. It is necessary to know the principles of fluid flow in order to design the plant. Gases are caused to flow through furnaces comprising mains and chambers of many sizes and shapes. Knowledge of the pressure variations in these systems is important in order that the furnace shall be so designed that a proper flow of gases is maintained; on the correct flow of the gases depends the heating of the furnace and the stock contained in it. Draught (see Chapter VI) is a special case of fluid flow. Convection heating is effected by fluid flow. The measurement of fluid flow is essential to the control of industrial operations. In this chapter, methods are given for the calculation of fluid flow with as simple an explanation as possible of the mathematical basis of these methods.

#### LIST OF SYMBOLS USED IN THIS CHAPTER

		Pronunciation of Greek letter.
Absolute viscosity—c.g.s. units (poises), f.p.s. units (no special name) .. ..		eta
Kinematic viscosity—c.g.s. units (stokes), f.p.s. units (no special name) .. ..		nu
Density of fluid—lb. per cubic foot ..	$\rho$	rho
Gravitational constant—32.2 ft./sec./sec.	$g$	
Static pressure—feet head of fluid ..	$x, h$	
Velocity of flow—feet per second. . .	$v$	
Coefficient of friction .. ..	$f, \zeta$	zeta
Frictional force per unit area of wetted surface .. ..		
Mechanical work done—foot-pounds ..	$w$	
Work done by expansion of a fluid—foot- pounds .. ..	$X$	
Pressure .. ..	$p$ (lb./sq. ft.) $P$ (inches w.g.)	
Volume .. ..	$V, Q$	
Diameter of pipe or conduit .. ..	$D$ (feet) $d$ (inches)	
Reynolds number .. ..	$Re$	
Length of pipe or conduit—feet .. ..	$L$	
Hydraulic mean depth (= sectional area ÷ wetted perimeter) .. ..	$M$	
Sp. gr. (air = 1) .. ..	$s$	
Coefficient of flow .. ..	$\alpha$	alpha

N.T.P., meaning normal temperature and pressure, refers to a gas at 32° F. and 29.92 inches bar., i.e. 0° C. and 760 mm. bar.

S.T.P., meaning standard temperature and pressure, refers to a gas at 60° F. and 30 inches bar.

## THE FLOW OF FLUIDS

### FLUID FLOW

#### THE MECHANICS OF FLOW

As in other branches of mechanics, the law of conservation of energy requires that the total energy in the system must remain constant.

The kinetics of fluid flow are best illustrated by reference to the fundamental basis upon which they have been worked out, namely Bernoulli's theorem. This may be illustrated in its simplest form by a liquid flowing through a pipe. The total energy possessed by the liquid at any point depends essentially on the algebraic sum of four factors, namely:—

Potential energy due to position + potential energy due to pressure + kinetic energy due to motion — energy lost as friction.

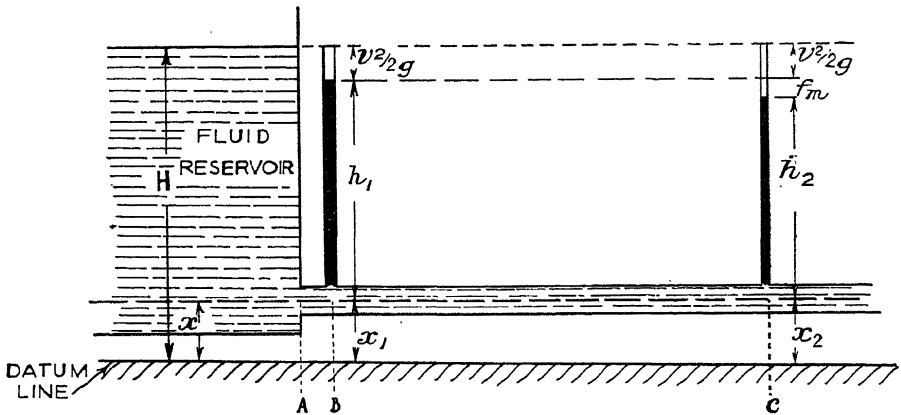


FIG. 54. Diagram illustrating Bernoulli's theorem of fluid flow.

If  $x$  is the height in feet above an arbitrary datum line (Fig. 54),

$h$  is the static pressure (measured in terms of a pressure gauge) expressed as feet head of the liquid under consideration.

$v$  is the velocity of flow of the liquid in feet per second.

$f_m$  is a measure of the friction losses.

$m$  is the mass of the liquid in lb./cu. ft.

$g$  is the gravitational constant = 32.2 ft./sec/sec.

Then by elementary mechanics:—

$$\begin{aligned} \text{mechanical energy} &= mgx + mgh + \frac{1}{2}mv^2 - : \\ &= mg \left( x + h + \frac{v^2}{2g} - : \right) \end{aligned}$$

and since the energy of the fluid in general terms =  $mgH$

where  $H$  is the effective head,

Fig. 54 indicates how the energy is distributed in practice. Here  $H$  is the total head available; part of this at any given height is due to the vertical distance from the (arbitrary) datum line. If the pipe is horizontal this distance  $x$  remains constant. As the liquid flows through the pipe, neglecting friction in the short length  $AB$ , part of the remaining energy  $H - x$ , is converted into the kinetic energy of motion and part remains as the static pressure  $h_1$ . Over the length  $BC$  if the pipe is of uniform bore, the velocity remains constant and so,

therefore, does the value of the velocity head  $v^2/2g$ . Frictional losses, however, reduce the static pressure, so that a pressure gauge indicates a drop in pressure between B and C of  $h_1 - h_2$  (i.e. of  $f_m$ ).

Bernoulli's theorem for gases is a little more complicated since the energy derived from compression or expansion of the gases may have to be taken into account. In a complete system, moreover, any work done on the fluid by a pump, etc., must be considered. The complete Bernoulli theorem for all fluids is given below. It is based, as stated above, on the principle that the total energy of the fluid at any one section of the system must equal its total energy at any other section. Thus :—

$$x_1 + h_1 + v_1^2/2g + W + X = x_2 + h_2 + v_2^2/2g - f_m$$

where the notation is as in Fig. 54

and- $W$  = work done mechanically, e.g. by a pump

$X$  = work done by expansion of the fluid =  $\int p dv = RT \log (V_2/V_1)$

$h_1, h_2 = p.V$ , when  $p$  is the pressure of the fluid and  $V$  is its volume under the conditions obtaining at the point of measurement.  $R$  is the gas constant and  $T$  the absolute temperature.

Thus,

$$x_1 + p_1V_1 + v_1^2/2g + W + X = x_2 + p_2V_2 + v_2^2/2g - f_m \quad . \quad . \quad (1)$$

### STREAMLINE AND TURBULENT FLOW

For low rates of flow in a pipe of given diameter  $D$ , fluids will travel in a streamline motion as shown in Fig. 55 (a). The individual particles flow in straight lines parallel to the axis of the pipe. The curve of velocity at any given distance from the sides of the pipe is parabolic in shape as shown in that figure, where the lengths of the straight lines is a measure of the relative velocities.

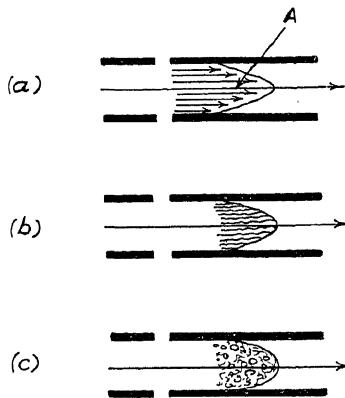


FIG. 55. Types of fluid flow.

In the same pipe when the velocity is further increased the straight-line flow becomes disturbed and a "wavy" or "transitional" type of flow is set up (Fig. 55 (b)).

A small additional increase in velocity causes the flow to become turbulent (Fig. 55 (c)). Here the straight-line flow is broken up by eddies and vortices. The velocity-distribution curve is different; it rises more sharply against the wall and is flatter in the central portion of the pipe.

The transitional velocity from one type of flow to another was investigated by Reynolds, and was found to depend on the viscosity, the diameter of the pipe and the velocity of flow of the gas. Viscosity is discussed in this chapter.

The relationship was expressed by the "Reynolds number,"  $Re$ , where

$$Re = \frac{vD\rho}{\eta} = \frac{vD}{\nu}$$

where  $v$  = velocity of gas  
 $D$  = diameter of pipe  
 $\rho$  = density of fluid

} These terms are defined on pp. 170 and 171.

Thus in c.g.s. units,

$$Re = \frac{v(\text{centimetres per second})D(\text{centimetres})\rho(\text{grams per cubic centimetre})}{\eta \text{ (poises)}}$$

and in f.p.s. units,

$$Re = \frac{\text{per second})D(\text{feet})\rho(\text{lb. per cubic foot})}{\eta \text{ (f.p.s. units)}}$$

Provided a consistent set of units is used, i.e. all in the c.g.s. system or in the f.p.s. system, the same value for  $Re$  is obtained by these formulæ. When  $Re$  is below about 2,000, the flow is streamline; between 2,000 and 3,000 it is transitional and above about 3,000, turbulent. In most industrial practice, when handling gases and liquids of low viscosity, flow is turbulent.

Although the foregoing discussion indicates the broad outlines of fluid flow, it is worth while examining the mechanism of flow more closely. At any average velocity of flow in a pipe, even when high enough to be turbulent, at the immediate surface of the pipe the velocity is very low, consequently the value of  $Re$  will be low and the flow is here streamline.

Likewise, a short distance farther from the pipe surface the flow will be transitional, whilst in the main body of fluid it will be turbulent.

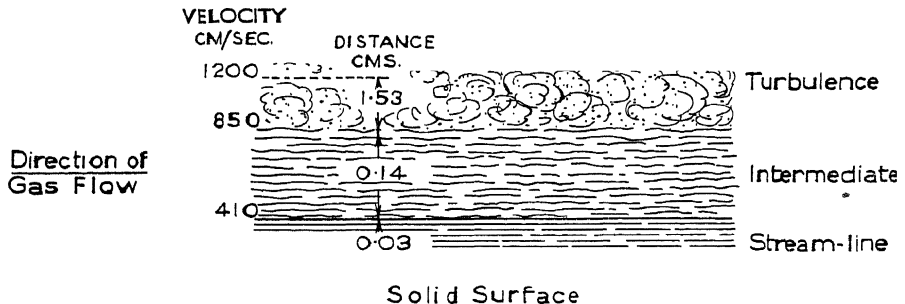


FIG. 56. Fluid flow near surfaces.

Thus experiments (Zijnen, Thesis, Delft, 1924) on air flowing parallel to a horizontal plate of glass gave the results shown in Fig. 56.

## EFFECT OF MECHANISM OF FLOW ON CONVECTION HEATING

The mechanism of flow influences the rate of heating by convection. In convection heating, it is necessary for a hot particle to come in contact with a cooler particle in order to share its heat energy with it. The cooler particle may be another gas or liquid particle, or it may be part of the solid heating wall, etc. Thus, if a heating fluid is circulating in a pipe, the objective must be to cause the heating fluid to give up its heat to the pipe through which the heat will be conducted.

A molecule of hot gas at the point A (Fig. 55 (a)) has little opportunity of coming in contact with the pipe. It is travelling in a streamline path and can thus only reach the side by a slow process of diffusion. It is likely that the heating medium will have left the apparatus before this happens. Consequently, the only means left for heat transfer from the particle at A is by thermal conduction. The thermal conductivity of gases and most liquids is very low, so that with streamline flow heat transmission must be slow.

A similarly situated particle in a fluid flowing turbulently will quickly be mixed with the remainder of the fluid which will thus tend to be at a uniform



temperature. Against the side of the pipe, however, there still persist the transitional and streamline layers, through which the rate of flow will be slow. These layers are thin, and consequently there is more opportunity for diffusion in them; they still have low thermal conductivity, but their small thickness enables heat to be transmitted more quickly. Thus, gaseous or fluid heating must always be slowed up by the presence of the streamline layer against the walls, but the thinner this layer, the more rapid the heating.

In the experiments of Zijnen previously mentioned, it was found that by increasing the velocity of the main stream of air from 13.1 to 78.7 feet per second, the thickness of the streamline layer was decreased from 0.087 to 0.018 centimetres. This indicates why higher velocities of flow give a greater rate of convection heating.

### VISCOSITY

A liquid which flows with difficulty is generally said to be viscous, and the property which determines the internal resistance to flow is its viscosity.

Resistance to fluid flow is caused by friction. Friction in solids is a force exerted between surfaces in contact. Fluids in motion are in contact with the surfaces of the pipes or ducts in which they move, but the layer of fluid in immediate contact with a surface is stationary so that the force acting between solid surfaces in contact does not operate in the same way between moving fluids and the surfaces by which they are bounded.

Particles of moving fluids do not move with uniform velocity throughout the whole area of a pipe or duct. Immediately against the surface there is the stationary layer, perhaps only one molecule thick. The work of Langmuir suggests that the first layer may be bound to the surface by physico-chemical forces.

Subsequent layers move with a velocity which increases with their distance from the surface. The work which must be expended in order to produce this relative motion represents frictional forces which come into play in the interior of the liquid. This force tends to retard the more rapidly moving layers and to accelerate those which are moving less quickly. The force required to cause the relative motion under standard conditions is known as viscosity.

The numerical value of viscosity at a given temperature is a specific physical property of a fluid, so that different fluids have different viscosities.

The magnitude of the viscosity in any fluid may be defined as ( $\alpha$ ) the force expressed in dynes acting on an area of 1 square centimetre necessary to produce a difference of velocity of 1 centimetre per second between a layer of fluid and a similar layer 1 centimetre distant. Thus in Fig. 57, if the layer of

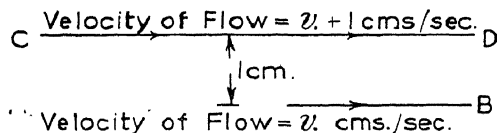


FIG. 57. Illustration of the principle of viscosity.

fluid AB moves with a velocity  $v$  centimetres per second, and the layer CD moves with a velocity of  $v + 1$  centimetres per second, the force in dynes per square centimetre acting on the layer CD just sufficient to cause this difference of motion is known as the coefficient of viscosity in c.g.s. units. This unit is known as the "poise."

( $\delta$ ) In f.p.s. units, the coefficient of viscosity is the force in poundals, which acting on an area of 1 square foot will produce a differential velocity of 1 foot

per second in two layers of fluid 1 foot apart. There is no name for the f.p.s. unit. Numerically,

Viscosity in poises  $\times 0.0672$  = viscosity in f.p.s. units.

Viscosity in f.p.s. units  $\times 14.88$  = viscosity in poises.

These units are known as "absolute" viscosity, and usually designated by the Greek letter,  $\eta$ .

In practical work it is often convenient to use another value known as "kinematic" viscosity commonly denominated by the Greek letter,  $\nu$ . In c.g.s. units this is known as the "stokes."

Kinematic viscosity =  $\frac{\text{absolute viscosity}}{\text{density}}$

which, in c.g.s. units =  $\frac{\text{poises}}{\text{grams per cubic centimetre}}$  stokes

and, in f.p.s. units =  $\frac{\text{f.p.s. units of viscosity}}{\text{lb. per cubic foot}}$  ft.-lb.-sec. units

kinematic viscosity in stokes  $\times 0.0010764$  = f.p.s. units

kinematic viscosity in f.p.s. units  $\times 929.03$  = stokes.

It is commonly appreciated that viscous liquids become thinner when heated. They flow more readily in this condition. This means in practice that the force required to overcome the internal friction becomes less. For liquids, therefore, viscosity becomes less as the temperature rises. In gases, however, the opposite occurs, and the viscosity becomes greater as the temperature rises. This fact is not always appreciated. Its practical significance is that the power required to move gases, e.g. in fans, becomes greater at higher temperatures. Thus in supplying preheated gases to burners not only does the greater frictional resistance produced by the expansion of the heated gases require to be overcome, but also the effect of the increased viscosity.

Viscosity changes rapidly with temperature. Thus kinematic viscosity must always be calculated with reference to the viscosity and density of the fluid at the temperature concerned. It is of great importance in all problems connected with the flow of fluids. At pressures reasonably near atmospheric, even up to, say, 100 lb. per square inch, viscosity may be regarded as independent of the pressure.

No general law can be given regarding the effect of temperature on the viscosity of liquids. Experimentally determined values must be taken. In Table 48 are given some values for water (Bingham and Jackson, *Bulletin, Bureau of Standards*, 1917, **14**, 75); intermediate values can be obtained by interpolation. Values for certain other liquids are to be found in Table 49.

TABLE 48. VISCOSITY OF WATER

Temperature		Absolute viscosity, $\eta$		Kinematic viscosity, $\nu$	
°C.	°F.	Poises	F.P.S. units	Stokes	F.P.S. units
0	32	0.01792	0.001204	0.01792	-00001930
10	50	0.01308	0878	0.01308	1407
15	59	0.01140	0766	0.01141	1228
20	68	0.01005	0675	0.01007	1084
30	86	0.00801	0538	0.00804	0866
50	123	0.00549	0369	0.00556	0599
70	158	0.00406	0273	0.00415	0447
100	212	0.00284	0191	0.00296	0319

TABLE 49. KINEMATIC VISCOSITIES—STOKES

Temperature		Petrol No. 1	Power alcohol	Motor benzole	Diesel oil	Heavy fuel oil
° C.	° F.					
0	32	0.0080	—	—	0.108	204
10	50	0.0074	0.015	0.0083	0.070	50
20	68	0.0068	0.012	0.0072	0.049	12
30	86	0.0062	0.010	0.0066	0.040	5

The viscosity of fuel gases is given in Tables 50 and 51.

The kinematic viscosity at any given temperature,  $t^\circ$  is

$$\frac{\text{absolute viscosity at } t^\circ}{\text{density at } t^\circ} \text{ i.e. } \frac{\eta_t}{\rho_t}$$

Both the viscosity and density must be expressed throughout in the same units system, e.g. in c.g.s. units or in f.p.s. units.

TABLE 50

ABSOLUTE VISCOSITY OF INDUSTRIAL FUEL GASES IN C.G.S. UNITS (POISES)

Calculated from the modified Sutherland formula,

$$\eta_t = \frac{\eta_0}{1 + t/C^1} \times \left( \frac{273}{273 + t} \right)^{1.75}$$

when  $\eta_t$  and  $\eta_0$  are viscosities at  $t^\circ$  C. and  $0^\circ$  C. respectively.

Type of gas	Blast furnace gas	Coke oven gas debenzolised	Continuous vertical retort gas (steaming)	Mechanical producer gas		Blue water gas	Steam	Air
				Coal	Coke			
Density at $0^\circ$ C. and 760 mm.	0.0013	0.00049	0.00062	0.00113	0.00116	0.000712	0.000804	0.001293
Temp.	$10^6 \times$ absolute viscosity in C.G.S. units							
$0^\circ$ C.	160	114	132	163	166	153	88	170
20	170	121	140	172	176	162	95	180
40	179	128	148	181	185	171	103	190
80	196	141	162	198	203	187	118	209
120	214	153	177	215	220	203	135	225
160	230	166	191	231	236	217	152	242
200	245	177	204	248	252	231	167	258
250	263	190	219	263	270	248	186	276
300	280	203	233	280	287	264	203	295
400	312	227	261	312	319	293	238	329
500	342	249	285	341	349	321	272	359
$C^1$	405	423	416	393	394	392	950	397

Densities in gm./c.c. at  $0^\circ$  C. and 760 mm.

To obtain values for density in lb./cu. ft. at  $32^\circ$  F. and 29.92 in. bar., multiply by 62.428.

To obtain value for viscosity in f.p.s. units, multiply by 0.0672.

TABLE 51. KINEMATIC VISCOSITY OF INDUSTRIAL FUEL GASES IN C.G.S. UNITS (STOKES) AT ATMOSPHERIC PRESSURE

Derived from values of absolute viscosity calculated according to the modified Sutherland formula by dividing by the density at the corresponding temperature.

Type of gas Temp.	Blast furnace gas	Coke oven gas debenzolised	Continuous vertical retort gas (steaming)	Mechanical producer gas		Blue water gas	Steam	Air
				Coal	Coke			
0° C.	0.123	0.233	0.213	0.144	0.143	0.217	0.109	0.131
20	0.141	0.264	0.242	0.164	0.162	0.246	0.128	0.152
40	0.159	0.298	0.274	0.184	0.182	0.276	0.147	0.172
80	0.197	0.371	0.338	0.228	0.226	0.342	0.191	0.212
120	0.238	0.449	0.411	0.275	0.272	0.412	0.240	0.254
160	0.282	0.535	0.489	0.325	0.322	0.488	0.295	0.300
200	0.328	0.622	0.570	0.378	0.374	0.567	0.356	0.346
250	0.390	0.740	0.676	0.448	0.444	0.672	0.436	0.413
300	0.455	0.866	0.789	0.523	0.517	0.783	0.528	0.480
400	0.595	1.136	1.037	0.681	0.676	1.023	0.730	0.628
500	0.749	1.437	1.300	0.857	0.850	1.280	0.963	0.787

To obtain value for viscosity in f.p.s. units multiply by 0.0010764.

### COEFFICIENT OF FRICTION

The viscosity of a fluid is a measure of its internal friction. Viscosity is not, however, the only cause of pressure loss. The eddies set up in turbulent flow are also responsible for frictional losses, and it is not surprising to find that there is a relationship between the coefficient of friction of fluid flow and the Reynolds number.

In order to determine the flow of a fluid it is thus necessary to calculate the Reynolds number for the conditions and from that to ascertain the coefficient of friction from experimentally determined data. The principles involved will now be discussed.

### EQUATIONS OF FLUID FLOW

The fundamental equation (derived from Bernouilli's theorem equation (1)) expressing the pressure drop along a horizontal pipe in which a fluid is flowing is :—

$$p = 8(F/\rho v^2) \cdot \frac{L}{D} \cdot \rho \frac{v^2}{2g} \quad \dots \dots \dots (2)$$

where  $p$  = pressure drop in lb./sq. ft.

$F$  = frictional force per unit area of wetted pipe surface

$\rho$  = density of fluid, lb./cu. ft.

$v$  = mean linear velocity of flow, ft./sec.

$L$  = length of pipe in feet

$D$  = diameter of pipe in feet

$g$  = 32.2 ft./sec./sec.

This equation is a more fundamental form of Fanning's equation to be discussed later.

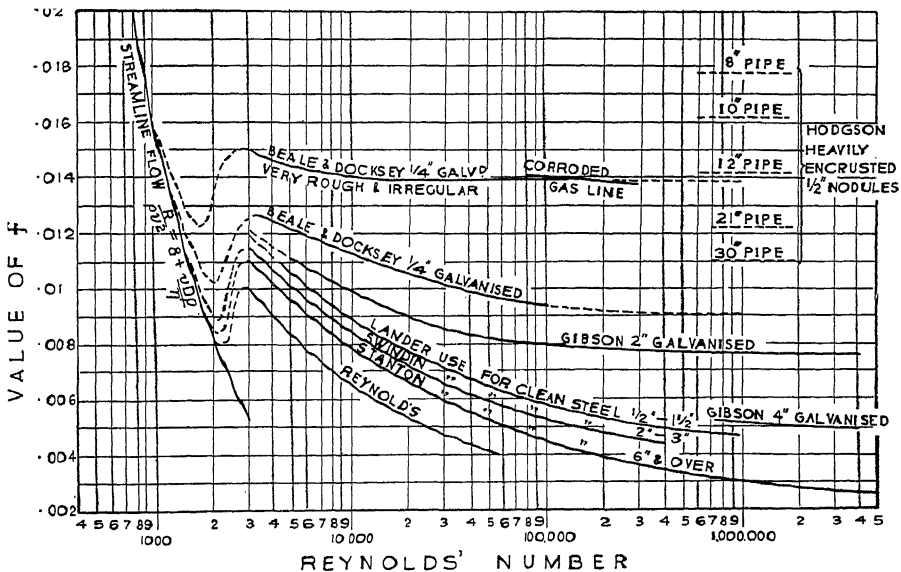
For *streamline flow*, Poiseuille's equation holds :—

$$32\eta Lv \text{ lb./sq. ft.} \dots \dots \dots (3)$$



TABLE 52. CALCULATED CRITICAL VELOCITIES IN STANDARD 2-INCH PIPE

Fluid	Temp., ° F.	Absolute pressure, atmospheres	Specific gravity, $s = \rho/62.3$	Relative viscosity, centi- poises	Critical velocity, ft./sec.
Hydrogen .. ..	70	1	0.0000834	0.0088	6.2
Air .. ..	70	1	0.0012	0.0184	0.904
Air .. ..	70	10	0.012	0.0184	0.0904
Steam .. ..	212	1	0.00060	0.0120	1.18
Steam .. ..	357	10	0.00524	0.0144	0.162
Water .. ..	68	Any	1.0	1.0	0.059
Refined oil .. ..	68	Any	0.90	10.0	0.65
California crude oil ..	68	Any	0.963	3,450	211

FIG. 58. Curves showing the relation between  $Re$  and  $f$ .

(Based on Fig. 8, page 64 of "Technical Data on Fuel," 4th edition.)

absolute pressure in lb. per square inch, and  $\rho_0$  and  $v_0$  are the values at 60° F. then under any temperature  $t$  °F. and pressure  $P$  lb. per square inch absolute.

$$\frac{\rho v^2}{2g} = 0.000439 \rho_0 v_0^2 (460 + t)/P \text{ lb./sq. ft.}$$

Under atmospheric pressure,  $P = 14.7$ .

Pressures in lb./sq. ft.  $\div 5.2 =$  inches water gauge.

The value of the friction factor,  $f$ , is taken from Fig. 58 which is a graphical representation of a large number of experimental results. In using this and similar tables it is to be noted that a common form of this method of expressing experimental results is to plot  $F/\rho v^2$  (cf. equation 1) against  $Re$ . Equation (2) can be resolved into

which is the same form as equation (5). Thus when  $F/\rho v^2$  is taken as the friction coefficient, its value must be doubled to accord with the value of  $f$  required in Fanning's equation.

In Fig. 58, the value of  $f = 2(F/\rho v^2)$ , and this value can be directly used in Fanning's equation.

*Formula (5) can be used for any form of liquid flow in circular pipes, and formula (4) when the pipes are not circular, by selecting the proper value of  $f$  from Fig. 58. Due correction must be made for bends, tees, valves, etc.; as given in Tables 53 and 54.*

It should be noted that the product  $v\rho$  in the expression for Reynolds' criterion is the mass-velocity. The data given in Fig. 58 shows how the frictional resistance varies with the roughness of the surfaces of the ducts.

Koo showed that for a limited range of  $Re$  from 5,000 to 200,000

$$f = 0.049/Re^{0.2}$$

For clean pipes of iron and steel, and flow in the turbulent range, the following equation holds within a deviation of  $\pm 10$  per cent. of the coefficient of friction,  $f$  :—

For badly corroded or scaled pipes, the value of  $f$  based on the observed pressure drop and the original diameter may rise to very high figures.

### ENLARGEMENT AND CONTRACTION LOSSES

In turbulent flow the loss in pressure is often due not only to the frictional effect of fluids flowing past the walls, but also to sudden changes in the cross-sectional area of the path.

When the section of the flue or pipe is suddenly *enlarged* a portion of the velocity pressure in the smaller section is transformed into static pressure, since the velocity is decreased. This is also accompanied by frictional loss due to impact of faster-moving particles on slow-moving particles.

The net result is a loss of pressure,  $h$ , measured as in equation (13) so that

$$h = (v_1 - v_2)^2/2g \quad \dots \dots \dots (13)$$

When the cross-section is suddenly *reduced*, the velocity is increased and part of the up-stream static pressure is transformed into velocity pressure in the smaller section; there is also a loss of pressure through impact. The overall loss,  $h$ , is given by

$$h = kv_2^2/2g$$

where  $v_2$  is the velocity in feet per second in the smaller section and  $k$  is a numerical factor which depends on the ratio of the smaller to the larger pipe area.

When this ratio = 1	0.8	0.6	0.4	0.2	0
$k = 0$	0.15	0.25	0.34	0.43	0.5

### EFFECT OF TEES, VALVES, ELBOWS AND JOINTS IN PIPE LINES

The effect of changes in direction due to tees or elbows is best allowed for by regarding the frictional effect as equivalent to a length of straight pipe. This length would be added on to  $L$  in the formulæ given in this chapter. Table 53 represents recommended practice.

Joints in pipe lines increase the friction loss, since in effect they accentuate the roughness of the interior surface. The friction coefficients given in Fig. 58 are for straight lengths of pipe without joints. All industrial pipes are jointed, of course, and the values of the coefficient should be increased as in Table 54 in accordance with the Reynolds number.

TABLE 53. FRICTION LOSS DUE TO BENDS, ELBOWS, VALVES AND TEES FOR CIRCULAR PIPE

This table applies to all fluids, but for turbulent motion only.

Type of bend	Diam. of pipe ins.	Equivalent length, expressed as diameters of additional straight pipe, i.e. L/D
90° bend .. .. .	$\frac{1}{2}$ -2 $\frac{1}{2}$	30
	3-6	40
	7-10	50
45° bends		
90° bend (long radius) }	1-3	15-20
90° elbow (sharp) .. .. .	—	50
90° curve, same inside diameter as pipe :—		
centre line radius = diam. of pipe ..	—	20
do. do. = 2-8 diameters ..	—	10
Tee, full-size branch .. .. .	1-4	60-90
(use higher value when fluid enters branch)		
Globe valves : .. .. .	$\frac{1}{2}$ -2 $\frac{1}{2}$	45
	3-6	60
	7-10	75
Gate valves—open .. .. .	—	7
Do. do. —Rises with successive closed positions, till at $\frac{1}{4}$ -closed it is	—	800

TABLE 54. PERCENTAGE INCREASE IN THE VALUE OF  $f$ 

Re	Rough joints	Smooth joints
	Per cent.	Per cent.
Up to 5,000.. ..	15	5
5,000-20,000 .. ..	10	3
Over 20,000.. ..	5	2

### FRICTION LOSS FOR FLOW OUTSIDE TUBES

*Parallel to Axis.* Generally the treatment already recommended suffices.

*Flow Normal to Axis.* The friction for turbulent flow of fluid normal to a bank of tubes,  $n$  rows deep, may be considered as due to  $n$  contractions and enlargements, thus :—

$$f = 4f'n \cdot v_{\max}^2$$

where  $v_{\max}$  = maximum velocity, feet per second

$w$  = mass of fluid, lb.

$g$  = gravitational constant

$f'$  has been shown by Andreas and Grimson to depend upon the arrangement.

For tubes in line

$$f' = \frac{(x_t - 1)}{x_t} \quad \text{for staggered tubes}$$

For staggered tubes

$$f' = \left( 0.23 + \frac{0.11}{(x_t - 1)^{1.08}} \right)^{-0.15}$$



Where  $x_1$  = ratio of the longitudinal pitch to tube diameter

$x_t$  = ratio of the transverse pitch to tube centre to centre distance.

$D$  = outside diameter in feet.

$n$  = No. of rows

$\eta$  = absolute viscosity expressed in f.p.s. units.

Values of  $\frac{vD\rho}{\eta}$  applicable are between 2,000 and 40,000.

### THE TRANSMISSION OF GASES

Gaseous flow requires special treatment if the pressure changes materially along the pipe.

#### LOW PRESSURE GAS TRANSMISSION

First, let it be supposed that the pressure does not change materially, so that the Fanning equation is directly applicable. It is convenient for practical reasons to use notation somewhat different from that of equations (4) and (5). Thus  $v$  will be replaced by  $Q$ , the quantity of gas flowing in cubic feet per hour at S.T.P.;  $D$  the diameter of the mains in feet by  $d$ , the diameter in inches;  $\rho$ , the density of the gas by  $s$ , its sp. gravity (compared with air as 1); and  $p$  (lb. per square foot) by  $P$  (inches w.g.).

Clearly,  $d = 12D$

and, since air weighs 0.076 lb./cu. ft. at S.T.P.

$$\rho = 0.076 s$$

$$P = p/5.2.$$

Consequently, the expression

becomes

$$P = \frac{4 \times 12 \times 0.076}{2g \times 5.2} \cdot fLs$$

$$= 0.01089 fLs \cdot v^2/d$$

But  $Q$  (cu. ft./hr.) = 3,600

$$= \frac{3,600}{4 \times 144}$$

whence  $v = Q/19.63 d^2$

Inserting this value in (7) and solving for  $Q$

If  $f$  be taken as 0.0065—a constant value for all conditions—and  $L$  be expressed in yards instead of feet, this formula becomes

which is Pole's well-known formula for the flow of gas in pipes that has for so long been used by the gas industry.

Pole's formula (9) is reasonably correct for low pressures up to about 6–8 inches w.g. It is less correct than (8), now known as Lacey's formula, even at these low pressures.

Lacey's formula (8) can be used up to about 3 lb. per square inch gauge pressure, the value of  $f$  being read from Fig. 58, as before,

## HIGH PRESSURE GAS TRANSMISSION

For high pressure gas transmission (see Unwin, "Treatise on Hydraulics," Chapter XI, and Lacey, *Trans. Inst. Gas E.*, 1927) the formula adopted in gas units is

$$Q = 2,675 \sqrt{\frac{d^5}{f s L} \cdot \frac{(p_1^2 - p_2^2)}{p_a^2}} \text{ cubic feet per hour at S.T.P.} \quad (9a)$$

where  $p_1$  and  $p_2$  = absolute pressure in lb./sq. in. (not per sq. ft.) at beginning and end of pipe

$p_a$  = absolute pressure of atmosphere in lb./sq. in. (generally 14.7 lb./sq. in.)

$L$  = length of main in feet.

Other formulæ have been proposed, but Unwin's is probably the soundest fundamentally.

## THE MEASUREMENT OF FLUID FLOW

The practical side of the measurement of gases, steam and liquids is discussed in Chapter X. An account is here given of the principles on which it is based.

From equation (1) it can be seen that if a pipe or duct in which a fluid is flowing is horizontal,  $x_1 = x_2$ .

Over a very short length,  $f$  may be neglected. If no work is done on the fluid,  $W = 0$ . For liquids,  $X = 0$ , and for gases, when the pressure drop is small,  $X$  can be neglected. Thus equation (1) becomes

whence

where  $h$  is the difference in head (expressed as feet head of the liquid or gas under consideration) between two points taken.

From this

## THE STANDARD ORIFICE

If an orifice be inserted in a pipe line as in Fig. 59, a pressure drop is caused

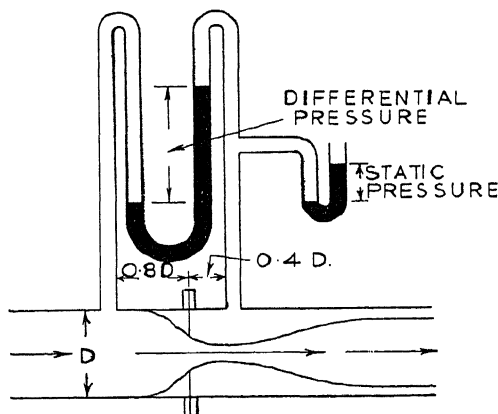


FIG. 59. Orifice plate and gauge.

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and in this way a portion of the static head  $h_1$  (Fig. 54) is converted into velocity head in the restricted section. The observed decrease in static pressure (obtained by a differential gauge) is caused by the increase in fluid velocity and theoretically the flow can be calculated as in equation (10).

In practice the flow is less than that given in equation (10), because this has been based on the assumptions that there is no friction loss and that the stream does not contract. As neither of these assumptions is correct, the discrepancy is allowed for by introducing a "coefficient of discharge,"  $\alpha$ , as in equation (11).

(11)

where  $v_1$  is the average up-stream velocity in feet per second

$v_2$  is the average velocity through the orifice in feet per second.

If a sharp-edged orifice be used and its diameter is large compared with the thickness of the plate,  $\alpha = 0.61$  (for exact values, see Chapter X).

If the diameter of the pipe be five times that of the orifice, the contraction of the stream, and therefore its velocity, is greatest at a distance down-stream  $4/10$ ths of the diameter of the pipe. By situating one measuring pipe at this point, and another not less than  $8/10$ ths diameter up-stream from the orifice, formula (11) simplifies to

$$v_2 = \alpha \sqrt{2gh} \quad . . . . . (12)$$

since  $v_1^2$  is negligible compared with  $v_2^2$ .

The sole disadvantage of this method lies in the fact that the suddenness of the velocity change causes a permanent loss of much of the observed drop in static pressure, which is converted into heat due to impact and internal friction. If the ratio of orifice diameter to the diameter of the pipe be 1 : 5, the loss is 93 per cent. of the static pressure drop. With a ratio 1 : 3, the loss is 85 per cent. and at 1 : 1.5, 55 per cent.

## THE VENTURI METER

If the orifice is streamlined and the change in section is gradual  $\alpha$  can become very nearly 1, say 0.97 or 0.99. Moreover, if the pipe section be gradually

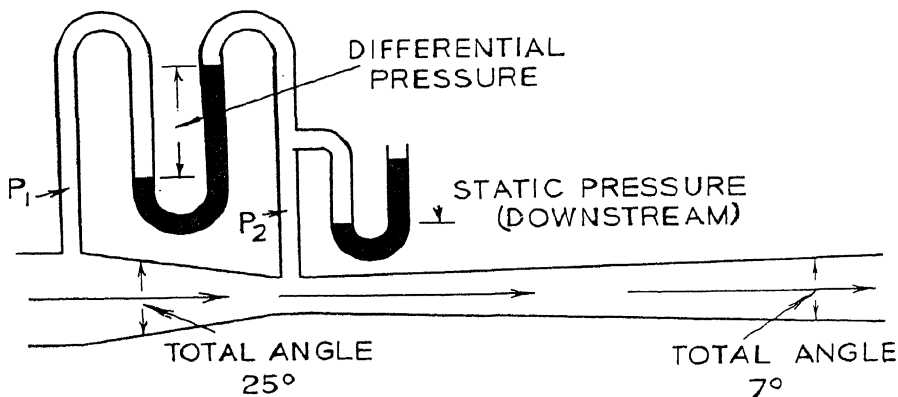


FIG. 60. Venturi meter.

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(instead of suddenly) enlarged to its former size, most of the increased velocity head is not lost, but is reconverted into static pressure. Under these conditions the orifice becomes a throat.

This arrangement and the general conditions required to make it effective are shown in Fig. 60. This is known as the Venturi meter. The equation of flow is equation (11). The permanent loss of static pressure with the arrangement shown in Fig. 60 is only 10–12 per cent. of the Venturi reading.

This meter is unsatisfactory with viscous liquids.

### THE PITOT TUBE

If, as in Fig. 61, one limb (*a*) of a differential pressure gauge is connected to an opening flush with the side of the main or pipe and the other (*b*) is connected

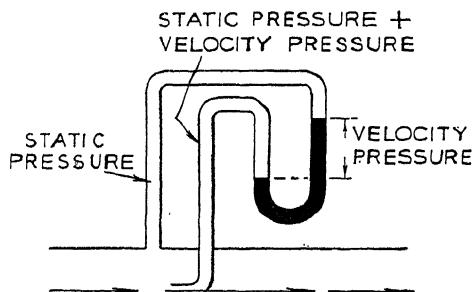


FIG. 61. Pitot tube.

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to a tube having its open end pointing against the gas stream, pipe *a* is subjected to the static pressure only and pipe *b* to the sum of the velocity and static pressures, so that the differential reading will indicate the velocity head.

The velocity head, as has been indicated in Fig. 54, is  $v^2/2g$ . Therefore if

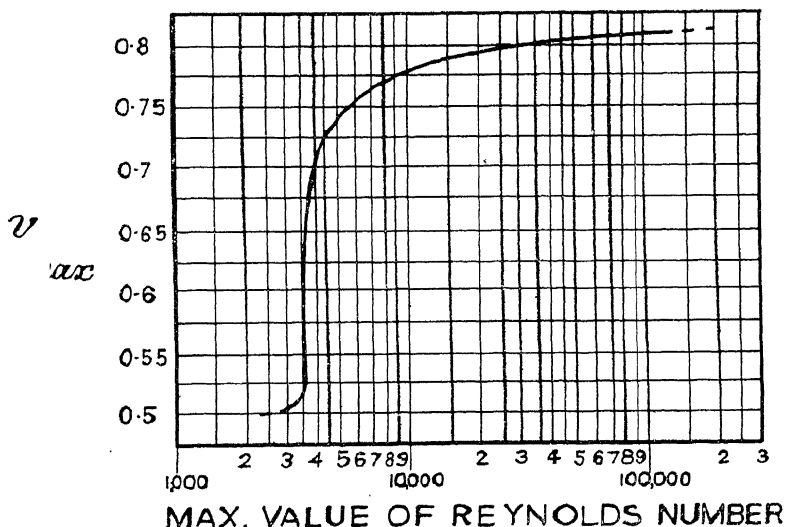


FIG. 62. Ratio of mean velocity to maximum velocity in circular pipes.

(From *Nat. Phys. Lab.*, 11, 1914.)

the reading of the differential gauge, expressed in feet of the fluid under consideration, is  $h$ ,

$$\begin{aligned} h &= v^2/2g \\ v &= \sqrt{2gh} \end{aligned} \quad \dots \quad (12a)$$

This measured velocity,  $v$ , is the velocity at the point where the tube facing the gas stream is situated. If this tube is placed along the centre line of the pipe, and there are no disturbing effects from bends, etc., the ratio of the average velocity throughout the pipe to the maximum velocity (at the centre line) is given by Fig. 62, which represents results obtained at the National Physical Laboratory in 1914. It will be seen that the ratio depends on the Reynolds number calculated for conditions at the point of measurement.

### THE MEASUREMENT OF FLUIDS IN DUCTS OTHER THAN CIRCULAR

In general, when it is required to measure flow in ducts that are not circular, the problem concerns hot gases. The Pitot tube is generally the most convenient method, the open end being placed, as before, on the central line of the flue. The calculation is made as above, but in place of taking the diameter of a circular tube for calculating the area of flow, the hydraulic mean depth,  $M$ , is used.

### THE DIMENSIONING OF STEAM PIPES

The size of steam pipes and mains can, of course, be calculated from Fanning's equation, but as this involves a knowledge of the viscosity of steam at possibly high pressures and temperatures, saturated or superheated, the usual practice is to adopt experimentally determined formulæ, necessarily of an empirical nature.

A formula stated to be accurate is due to Dr. Wierz of Charlottenburg. This formula as modified for English units is given below, the various numbers contained in it being derived from his experimental results:—

$$\frac{p_2^{1.9375} - p_1^{1.9375}}{L} = 0.00012 \frac{Q^{1.853}}{d^{4.987}}$$

where,

$p_1$  and  $p_2$  are the final and initial pressures in a run of pipe in lb./sq. in. absolute.

$L$  is the equivalent length in feet of the pipe after the resistance of fittings has been taken into account,

$d$  is the diameter of the pipe in inches, and

$Q$  is the quantity of steam passing in lb./hr.

The correct dimensioning of steam pipes and mains is, of course, a matter of considerable importance, as if the pipes are too small an unduly high proportion of the pressure of the steam is used in overcoming the resistance.

Superheated steam can be passed through the pipes at about 50 per cent. higher velocity than saturated steam.

### STRATIFICATION

The subject of stratification has been neglected in technical literature, but is an important factor in many problems of heating and combustion. The mathematical treatment of the flow of fluids in pipes presupposes that when the flow is turbulent there is no stratification. In pipes and flues of 12 inches diameter this is not necessarily true universally and in wide pipes and flues it may be very far from true.

The velocity over even a small pipe is never uniform as has been shown earlier in this chapter, but the velocity over a large pipe or flue completely filled with the flowing gas (and especially if the flow is turbulent) may be

virtually uniform except at the edges ; on the other hand there may be wide differences in velocity, temperature and composition over the area of the flue.

As a result of stratification, stock in furnaces may be immersed in gases of a temperature different from the average that was anticipated ; measurements of temperature in flues may be very misleading unless the whole cross-section of the flue is explored ; and the  $\text{CO}_2$  content of the flue gases may vary in different parts of the flue.

In large flues two effects are always operative, the natural buoyancy of the gas, which tends to give separation of the heavier and lighter constituents in so far as such diffusion is not interfered with by turbulence, and the setting

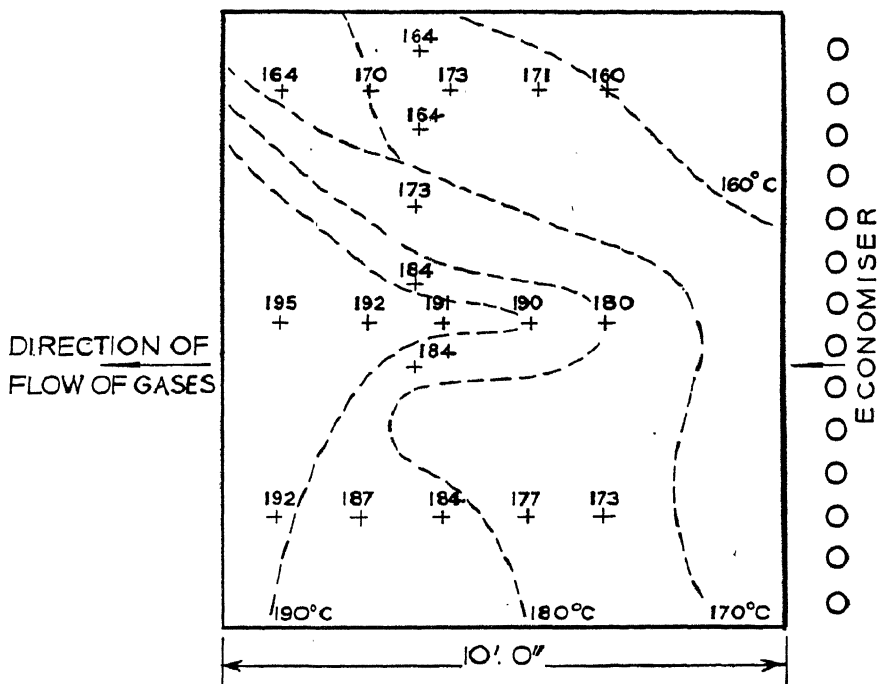


FIG. 63A. Stratification of gases at an economiser outlet.

up of secondary effects due either to heating or cooling of the gases by means of the flue walls. Attempts have been made to correlate the depth of a stream of hot gas flowing into a large duct from a smaller one on the basis of applying the analogy of water flowing over a weir, conditions being inverted when hot gas flows into the colder. For a discussion of the subject the reader is referred to Groume-Grjmailo's work in the "Flow of Gases in Furnaces" (English translation by A. D. Williams).

In practice the conditions of flow in large flues are complicated and the general formulæ worked out for smaller flues do not necessarily apply.

Stratification may also arise from other causes :—

- (1) Local cooling, especially if the bottom of a flue be cooler than the top.
- (2) Inleakage of air causing in effect two converging streams of different temperature and composition.

- (3) Converging gas streams at different temperatures and of different compositions meeting in a common flue.
- (4) Jet effects introduced by high velocity streams entering those at lower velocity.

These causes are likely to set up differences in temperature and composition.

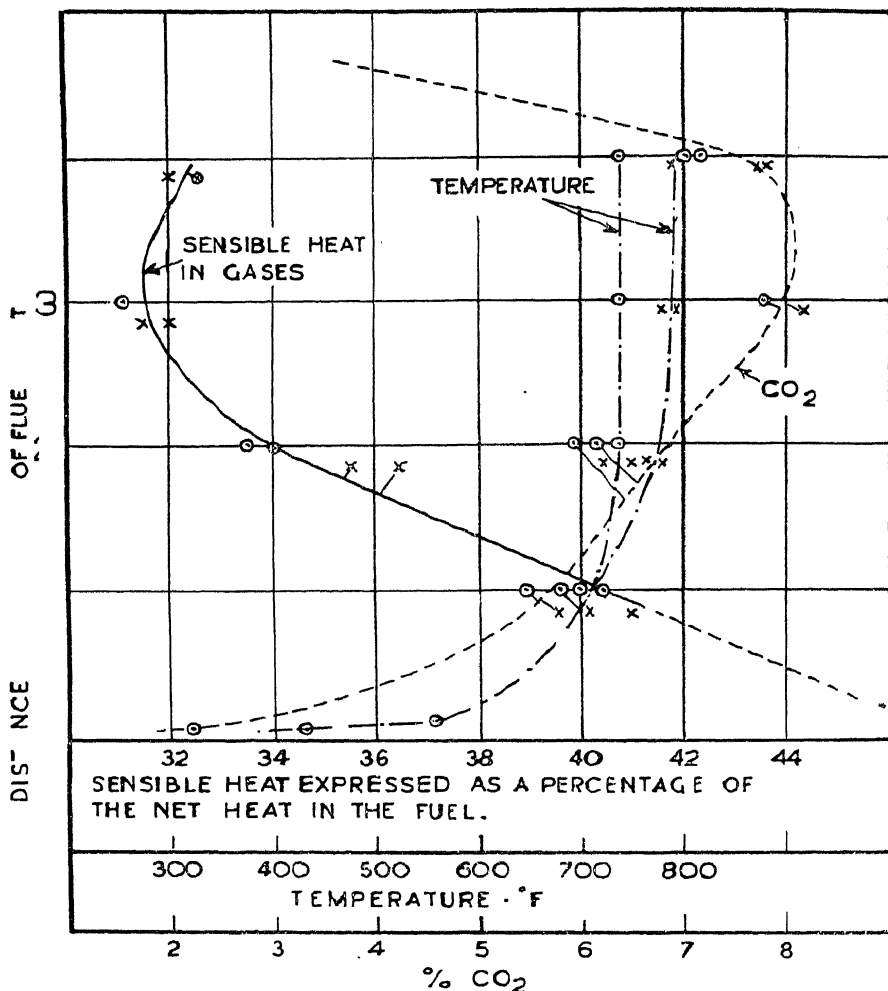


FIG. 63B. Stratification of gases in a boiler flue.

If the gas stream completely fills the flue and the flow is highly turbulent (i.e. Reynolds number high) the differences soon disappear, but if the Reynolds number is low and particularly if the gas stream does not fill the flue, the stratification may persist indefinitely.

Fig. 63a shows the temperature distribution in a boiler flue at the outlet from the economiser with the deflectors open and indicates that a single tem-

perature taken at this position would be quite unreliable as an indication of the true temperature of the gases. Fig. 63b shows an even more striking example of differences in temperature and  $\text{CO}_2$  content occurring in a boiler side flue 5 feet deep by 1 foot 6 inches wide, the temperatures and  $\text{CO}_2$  contents being taken from top to bottom.

The maximum content of sensible heat in the gases flowing in the flue is shown to occur towards the top of the flue, and it is apparent that in this zone there is a greater flow of the products of combustion than in the lower parts of the flue. The temperatures were measured with bare thermocouples of 18G, and would therefore indicate a temperature more nearly approaching the true temperature of the gas than would be found with the usual type of rod pyrometers, whose readings are influenced by radiation from the adjacent hot surfaces of the flue.

Two further examples may be cited. A main chimney flue for a coke oven plant fired by blast furnace gas showed the following variations :—

Depth from top of flue	Temperature	Depth	Analyses	
			$\text{CO}_2$	$\text{O}_2$
6 ins.	317 °C.			
1 ft. 6 ins.	421 °C.	1 ft.	19.8%	4.2%
2 ft. 6 ins.	317 °C.			
3 ft. 6 ins.	301 °C.	3 ft.	19.0%	4.6%
4 ft. 6 ins.	290 °C.			
5 ft. 6 ins.	286 °C.	5 ft.	19.6%	4.0%
6 ft. 6 ins.	283 °C.			
7 ft. 6 ins.	281 °C.	7 ft.	19.0%	4.6%

Flue depth = 8 feet.

Temperature recorder was at 3 feet position recording 309° C.

True average temperature = 300° C.

The cold velocity was about 10 feet per second or the hot velocity about 20 feet per second.

An example of stratification due to converging streams of flue gases of different composition derived from coke oven gas (combined velocity at 60° F. = 10 feet per second) is as follows :—

Position	Bottom	4 ins. up	8 ins. up.	12 ins. up	16 ins. up (top)
$\text{CO}_2\%$ ..	6.5%	6.8%	8.8%	7.9%	6.0%
Temperature	400° C.	445° C.	498° C	490° C.	420° C.

### THE FLOW OF GASES IN FURNACES

A natural draught furnace system consists of a series of communicating chambers open at two points to the atmosphere, through which gas and air are drawn as a result of the liberation of heat by combustion at some point within the system.

The flow of gases in existing furnaces can be studied by taking the pressure and temperature of the gases at different points. When designing new furnaces the dimensions and shape of the chambers and intercommunicating passages or flues must be fixed partly by the requirements of the process and partly by known data on the flow of gases.



The first step in the aerodynamic design of a furnace is to estimate as fully as possible the force available to bring about the required gas flow. This will be the algebraic sum of a series of chimney effects and the initial pressure:—

Chimney effect where hot gas rises — chimney effect where hot gas descends  
— resistances in the system + initial pressure.

Chimney effects have been dealt with in some detail in Chapter VI to which readers are referred for methods of calculation. It is necessary here to distinguish between the chimney effect of a rising flue or chamber and the effect on the system of the main chimney. Boilers and those furnaces which have no working doors or openings for the charging of material to be heated operate by the draught provided. This creates the motive power necessary for moving the air through the furnace and subsequent flues.

In furnaces which have working doors to their heating chambers, e.g. reverberatory furnaces, melting furnaces, puddling furnaces, the main chimney is nothing more than a means of removing the products of combustion. The draught effect of the main chimney makes itself felt at some arbitrarily fixed point in the system. Thus a furnace may operate with a slight pressure in the furnace chamber and following this chamber there will be a section of zero pressure from which the chimney operates to remove the products of combustion.

## BUOYANCY

If a closed vertical tower contains a gas lighter than air, it can be shown that the buoyancy effect of the gases sets up a difference in pressure between the top and bottom of the tower equal to the difference in weights of 1 cubic foot of air and of the lighter gas multiplied by the height of the column,  
i.e. 
$$P_2 = P_1 - P_2 = (w_a - w_g)H \text{ lb./sq. ft.} \quad (14)$$

where  $p_1$  and  $p_2$  are the gauge pressures at the top and bottom of the tower respectively in lb./sq. ft.

$w_a$  and  $w_g$  are the densities of the outside air and the gas in lb./cu. ft. under the conditions prevailing.

$H$  is the height of the tower in feet.

The value obtained by (14) divided by 5.2 gives the pressure difference in inches w.g.

If the top of the tower is open to the atmosphere the system is in effect a chimney with the damper closed. Clearly  $p_1$ , the pressure difference between the tower and the atmosphere, = 0. Equation (14) becomes

$$p_2 = - (w_a - w_g)H \quad (15)$$

$p_2$  being the measured static suction at the chimney base. Similarly if the bottom were open and the top closed,  $p_2 = 0$  and

$$p_1 = (w_a - w_g)H \quad (16)$$

If both bottom and top were open and the relative densities of air and gas were maintained—this being the condition met in practice—there would be a flow of gas up the chimney arising from exactly the same cause, but part of the static pressure difference,  $p_1 - p_2$ , would be converted into velocity head and part would be lost in friction, the net effort being given by Fanning's equation (equation (4)).

## EFFECT OF BENDS AND OBSTRUCTIONS

In furnace draughting the conditions of flow differ to some extent from the flow in tubes. It is found in the large flues and ducts in furnaces carrying hot gases which are radiating to a colder surface that a more complex system of

flow occurs. The variation in conditions of flow arises mainly from the separation of hot and cold layers of gas in the furnaces, the effect of burners and the location of flue offtakes.

### PRESSURE DROP IN FURNACES

It is of considerable interest to be able to calculate the pressure drop through a furnace or similar structure, and although the problem is essentially one of design, ability to know when resistances are abnormal is often of considerable value to the operator. The best method of describing how this can be done is to give an example. The following calculations are due to Dr. A. H. Leckie of the Iron and Steel Industrial Research Council.

In designing any furnace the resistance to gas flow through the system must be calculated to ensure that the necessary quantities of gas and air will enter the furnace with the pressure available and to calculate the draught necessary to exhaust the waste gases and infiltrated air.

In many furnaces, particularly those of the reversing regenerative type such as glass tanks or open hearth steel melting furnaces, most of the pressure drop is caused by the numerous bends and changes of section, and is due in only a minor degree to wall friction. In addition, substantial pressure changes are caused by the variations in vertical level through the furnace system, and the gas temperature is constantly changing. Therefore, although the pressure loss through a furnace may be evaluated by the Fanning equation (equation (4)), the numerous determinations of equivalent length, hydraulic mean depth, and Reynolds number involved make the calculation complex and it is preferable to use the alternative method outlined below.\*

The total static pressure change through a furnace system ( $P$ ) is divisible into two components, that due to eddying caused by bends, changes of section, and wall friction, which is proportional to the square of the N.T.P. velocity of the gas and is denoted here by  $p_v$ , and that due to changes in vertical level (buoyancy changes) denoted here by  $p_s$ , which is independent of the gas velocity. In Fanning's equation (equation (4)), the factor  $\rho v^2/2g$  is the dynamic or kinetic energy of 1 actual cubic foot of gas under the conditions in the flue; it is commonly termed the "velocity head." If  $v_o$  and  $\rho_o$  are the velocity (feet per second) and the density (lb. per cubic foot) at N.T.P., then at  $t^\circ \text{C}$ .

$$\rho_o v_o^2 \text{ becomes } \frac{\rho_o (273)}{(273 + t)} \times$$

$$\text{or } \frac{\rho_o v_o^2 (273 + t)}{273}$$

$g$  is unaffected by the temperature and thus at any temperature  $t^\circ \text{C}$ ,  $\rho v^2/2g$  may be written in terms of  $\rho_o$  and  $v_o$ ; thus for air the expression becomes

$$\frac{0.0805}{5.2 \times 64.4} \times \frac{v_o^2 (273 + t)}{273}$$

or  $8.80 \times 10^{-7} v_o^2 (273 + t)$  inches w.g.

and for any gas of specific gravity  $s$  (air = 1)

$$8.80 \times 10^{-7} s v_o^2 (273 + t) \text{ inches w.g.} \quad . \quad . \quad . \quad (17)$$

At every bend and change of section a certain fraction ( $f_1$ ) of this energy is lost due to eddying, and at the walls a fraction  $f_2$  is lost due to wall friction. Hence the total pressure loss,  $p_v$ , due to eddying and wall friction is given by

$$p_v = (f_1 + f_2) p_o \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

\* For a fuller description of this method of calculating furnaces see H. Etherington's "Modern Furnace Technology," pp. 124-190 (C. Griffin & Co. Ltd.).

and the total pressure (or draught) loss given by

(19)

The choice of sign depends on whether buoyancy assists or opposes gas flow in the direction required.

The value of  $p_1 - p_2$  may be calculated from equation (14) since

$$p_1 - p_2 = (w_a - w_g)H = p_s$$

where  $H$  is the height of the column of hot gas in feet. For air, for which  $w_a$  at atmospheric temperature may be taken as 0.0765 lb. per cubic foot

$$\begin{aligned} p_1 - p_2 &= \frac{(0.0765 - 0.0765s)H}{5.2} \\ &= 0.0147H(1 - s) \text{ inches w.g.} \end{aligned}$$

And at  $t^\circ \text{C.}$ , when  $t_a$  = atmospheric temperature in  $^\circ \text{C.}$ ,

$$p_1 - p_2 = 0.0147H \left( 1 - \frac{t_a}{t_a + 273} \right) \quad (20)$$

The crux of the pressure calculation is the selection of the correct values of  $f_1$  and (to a lesser extent)  $f_2$ . (This is, of course, a similar problem to the determination of equivalent length in Table 53.)

#### DETERMINATION OF $f_2$

The value of  $f_2$  may be disposed of briefly (it is the value of  $f \frac{L}{M}$  in the Fanning equation). Normally it is necessary to calculate the Reynolds number and the hydraulic mean depth, and so obtain the appropriate value after applying a correction for the nature of the surface of the duct. However, most furnace passages are constructed of brick, the surface of which becomes very rough after short use, and accurate determination of  $f_2$  is not possible. Since friction plays a relatively unimportant part in the total pressure drop, it is sufficient to consider only average flow conditions and to determine  $f_2$  from the dimensions of the flue only. For a fuller justification of this see Etherington (*loc. cit.* p. 155), but for ordinary works' calculation  $f_2$  may be determined as

$$f_2 = 0.0125 \frac{LC}{A} \quad (21)$$

where  $L$  is the length of the flue in feet.

$C$  is the cross-sectional perimeter of the flue in feet.

$A$  is the cross-sectional area of the flue in square feet.

#### DETERMINATION OF $f_1$

Values of  $f_1$  for bends and changes of section are listed by various authorities. Unfortunately the recommended values vary greatly, but those given below are what may be considered the best average as far as is known at present. (See Fig. 64.) Where there is a change in the N.T.P. velocity (i.e. velocity of the gas assumed measured at N.T.P.) the  $f$  value should be applied to the higher N.T.P. velocity.

(a) Right angle bend :— $f_1 = 1.5$ .

(b) Right angle bend made from T-piece by blanking off one short limb :— $f_1 = 2.0$ .

(c)  $45^\circ$  bend :— $f_1 = 0.5$ .

(d) Rounded bends :—If  $R > 5D$ ,  $f_1 = 0$ ; if  $R < 5D$ ,  $f_1$  increases up to the full 1.5 value for a  $90^\circ$  bend, according to  $R/D$ . ( $R$  = mean radius of bend,  $D$  = diameter of duct.)

- (e) Ultra-sharp bends such as are found between uptakes and ports:— $f_1$  may be 2.0 or more if the inside of the bend is not rounded off, but most bends of this type have the inside corner cut off to some extent either by design or through erosion after use when the  $f_1$  value is lower.
- (f) Entry to passage from large chamber or from atmosphere:— $f_1 = 2.0$ .
- (g) Exit from passage to large chamber or to atmosphere:— $f_1 = 1.0$ .

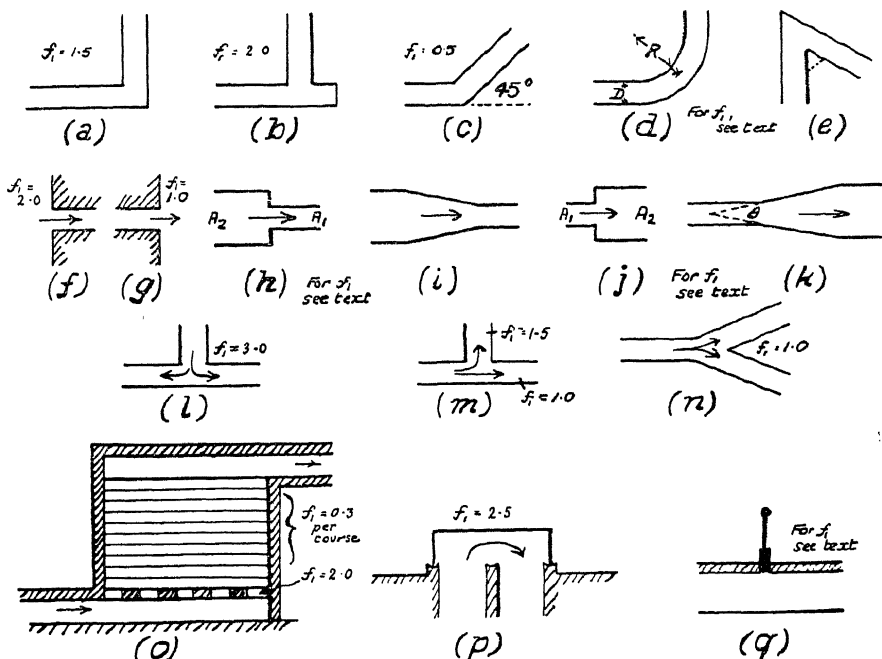


FIG. 64. Types of bends, changes of section, etc., commonly encountered in furnace systems.

- (h) Sharp-edged contraction:— $f_1$  depends on the ratio of the areas. Standard curves showing  $f_1$  plotted against this ratio are given in text-books (e.g. Etherington, p. 148), but in practice it is found that the  $f_1$  values so obtained are too low and the value used should be about three times that given by the curve. The recommended values of  $f_1$  are as follows:—

Area ratio ( $A_1/A_2$ ) = 0.1	0.2	0.4	0.6	0.8	1.0
$f_1 = 1.08$	1.02	0.81	0.5	0.16	0

- (i) Gradual narrowing:— $f_1$  very small.

- (j) Sharp-edged enlargement:— $f_1 = \left(\frac{A_2 - A_1}{A_2}\right)^2$  where  $A_1$  is upstream and  $A_2$  is downstream area.

- (k) Gradual enlargement:— $f_1 = k \cdot \left(\frac{A_2 - A_1}{A_2}\right)^2$ . According to Spiers'

"Technical Data on Fuel," 4th edition, p. 74,  $k$  has the following values for varying values for a gradual enlargement having a total angle of  $\theta^\circ$ :—

8	14	20	30	45	60	90–180 degrees
15	0.25	0.45	0.7	0.95	1.1	1.0

- (l) Subdivision of stream :— $f_1 = 3.0$ .
- (m) Subdivision of stream :— $f_1 = 1.0$  or  $1.5$  as shown.
- (n) Subdivision of stream :— $f_1 = 1.0$ .
- (o) Chequers :—For entry to chequers  $f_1 = 2.0$ . For the chequer courses themselves the  $f_1$  \* value of  $0.3$  per course of bricks for unstaggered patterns is suggested by Etherington ("Modern Furnace Technology," p. 144) and this seems to work well in practice. However, it must be applied to the velocity in the chequers and in calculating this it should be borne in mind that the theoretical free area of cross-section is seldom effective. The effective free area seems to be  $\frac{1}{2}$  to  $\frac{2}{3}$  of the theoretical free area according to the layout.
- (p) Reversing valves :—For types such as the butterfly, "Dyblie," "Lake," or "Forter,"  $f_1$  is approximately  $2.5$ .
- (q) Dampers :—If damper is fully open  $f_1$  may be taken as  $0$ ; if partly closed it should be treated as a contraction followed by an enlargement, the values from (h) and (j) being added.

The  $f_1$  values for any particular section may be added together, although in practice, where two items such as bend and change of section coincide or follow in quick succession, the total  $f_1$  value is somewhat less than the sum of the two separate components. However, the reduction in total  $f_1$  caused by proximity can be ignored as it is preferable to assume a slightly high value as this gives a factor of safety in designing the passages.

The principles and data just given will now be applied to a furnace in order to illustrate their application.

A regenerative slab heating furnace is to be constructed to the lines and dimensions sketched in Fig. 65. The producer gas flow to the furnace is  $137,000$  cubic feet per hour (measured at N.T.P.), the gas enters at  $500^\circ\text{C}$ . and is preheated to  $800^\circ\text{C}$ . The air required is  $140,000$  cubic feet per hour† (or  $38.9$  cubic feet per second) (measured at N.T.P.) and enters from the atmosphere, being preheated in the chequers to  $900^\circ\text{C}$ . It is required to know

- (a) the gas pressure required at the reversing valve.
- (b) whether an air fan is necessary,
- (c) the draught required in the stack flue.

The pressure balances are most conveniently calculated in tabular form. Table 55 is that for ingoing gas and Table 56 that for ingoing air. The furnace is most conveniently divided into the sections listed in Column 1. Column 2 shows the average temperature likely to be encountered in each section. Columns 3, 4 and 5 show the length, cross-sectional perimeter and cross-sectional area of the passages concerned. Column 6 gives the calculated N.T.P. velocity in that particular section. Column 7 shows the velocity head  $p_v$  calculated from equation (17). Columns 8–10 give the  $f$  values; reasons for the selection of these values are given in the Notes to Tables. Finally, the resulting value of  $p_v$  is tabulated in Column 11.  $p_v$  is separately tabulated below the  $p_v$  table.

It will be noted that in several parts of the furnace values of  $L$ ,  $C$  and  $f_2$  are not tabulated. This is because  $L$  is so small that  $f_2$  is negligible and knowledge of these quantities is not required.

Table 57 is the draught balance for the waste gases. Since the dimensions  $L$ ,  $C$  and  $A$  are already listed in Tables 55 and 56 these columns are not repeated. In calculating the resistance for the outgoing gases a slight complication is introduced by division of the waste gases between the gas and air chequers.

\* The  $f_2$  value for chequers is included in this.

† This quantity of air is less than the theoretical for complete combustion, but is the required inflow so that after allowing for infiltration, a slightly reducing atmosphere is maintained in the working chamber.

This is surmounted by assuming an arbitrary partition, say 30 per cent. through the gas chequer and 70 per cent. through the air chequer. The resistances will then usually work out to a different value for each path. It is known that the total pressure change through each path must be the same, and that  $p_v$  depends on the square of the gas velocity while the buoyancy effect,  $p_s$ , does not change with velocity. Hence, if  $x$  is the true partition, and  $p_v$  the eddy losses calculated on a basis of 30 per cent. through the gas chequers, we have :—

$$\left(\frac{x}{0.3}\right)^2 p_v + p_s = \left(\frac{1-x}{0.7}\right)^2 p_v + p_s \quad \dots \quad (22)$$

Thus  $x$  is determined and either term above will give the true pressure change to be expected over the divided path.

In the present example, Table 57 shows that  $p_v$  for the waste gases passing through the gas system is .9046 inches w.g., and for those passing through the air system is 1.5952 inches w.g., assuming a 30 per cent. partition through the gas system.  $p_s$  is .2242 and .2547 for the gas and air systems respectively. Hence :

$$\left(\frac{x}{0.3}\right)^2 \times .9046 + .2242 = \dots \times 1.5952 + .2547, \text{ and } x = .365.$$

Hence, under the specified conditions, about 36.5 per cent. of the outgoing waste gases will pass through the gas regenerative system and the draught required in the stack flue is given by putting  $x = .365$  in either term of the above equation, the result being 1.57 inches w.g.

Hence calculation of the pressure balances shows that :—

- (1) The gas pressure required at the reversing valve is 1.22 inches (from Table 55).
- (2) An air fan to give 0.17 inches w.g. at the entrance to the air reversing valve will be necessary (from Table 56).
- (3) A stack to give 1.57 inches draught in the stack flue is required.

Table 56 shows that most of the resistance to the ingoing air is caused by the uptakes. If it is desired to avoid the installation of an air fan, and to run the furnace on "natural air"  $p_s$  for the ingoing air must be greater than  $p_v$ , therefore the size of the uptakes should be increased until  $p_v$  is reduced by the required amount.

#### NOTES APPLICABLE TO f VALUES IN TABLES 55-57\*

- 1 Perimeter of each passage.
- 2 Total area of passages.
- 3 Two-thirds of the theoretical free area. In a less favourable layout, such as an open hearth furnace provided with slag-pockets, half the theoretical free area should be used.
- 4 Velocity at entry. This gradually decreases along the passage as the gas enters the chequers.
- 5 Entry from gas box,  $f_1 = 2.0$ ; 90° bend at bottom,  $f_1 = 1.5$ ; enlargement to flue,  $f_1 = 0.5$ . Total 4.0.
- 6 Two slightly rounded 45° bends at 0.3 each (instead of 0.5),  $f_1 = 0.6$ .
- 7 For contraction from 10.75 to 7.5 square feet standard curve shows  $f_1 = 0.1$ , 3 times 0.1 = 0.3. Division into 3 passages,  $f_1 = 2.0$ . Total 2.3.
- 8 Entry to chequer,  $f_1 = 2.0$ ; 25 courses at 0.3, 7.5; emergence from chequer to collecting space at top, 1.0. Total 10.5. The  $f_2$  value for chequers is included in the assumed 0.3 per course of brick.

\* The reference numbers in brackets against certain values in Tables 55 to 57 refer to items 1 to 24 of these notes.

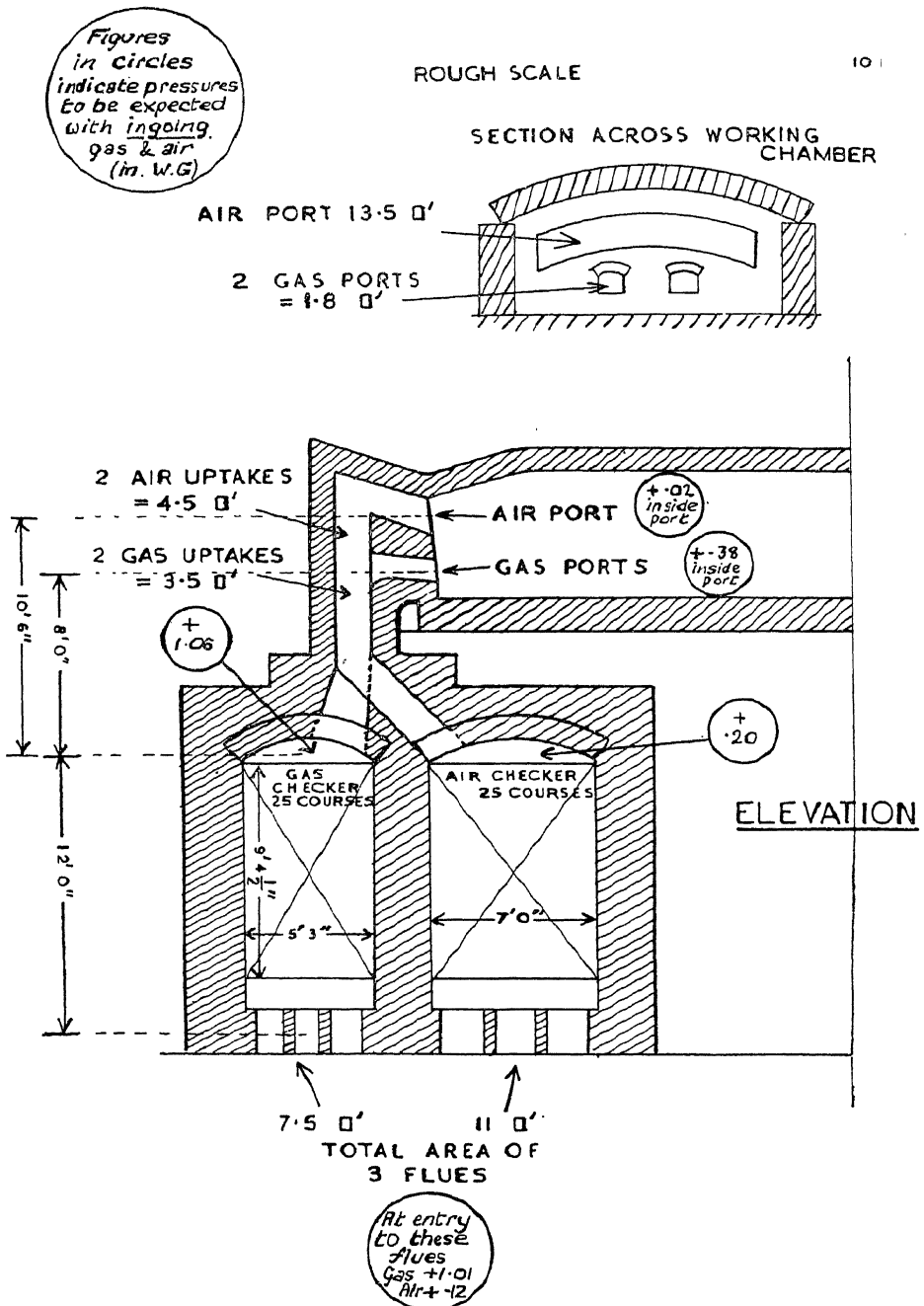


FIG. 65. Layout of regenerative heating furnace, to illustrate the method of calculating the pressure balance.

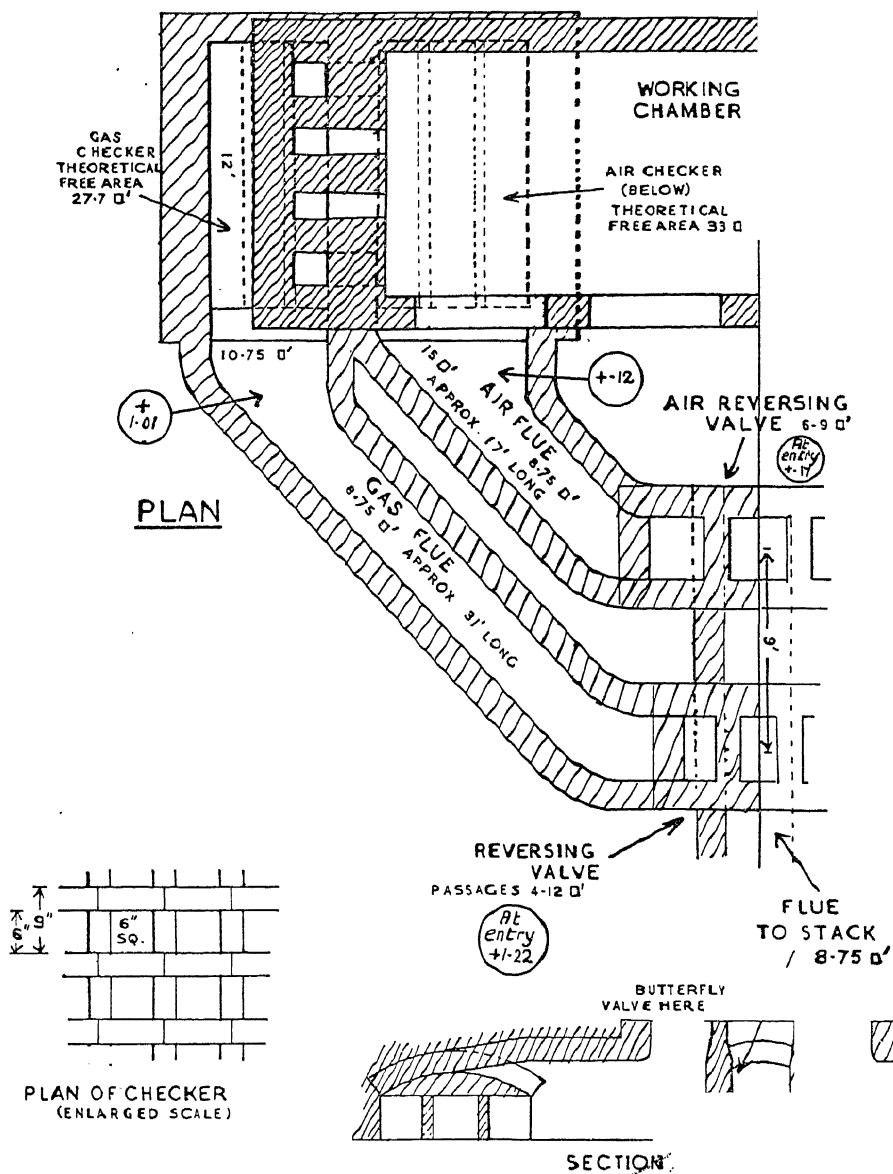


FIG. 65 (cont.).



- (9) The  $f_1$  value of 2.0 for the entry to the uptakes is applied to the velocity in the enlarged entry section. The gradual narrowing afterwards does not occasion any further  $f_1$  value, but the  $f_2$  value is applied to the velocity in the 3.5 square feet section which covers most of the length.
- (10) Somewhat rounded bend into port—assumed  $f_1 = 1.5$ . Assume 0.2 for the contraction on entering port, which is made rather gradual by the slight rounding of the bend. Loss of velocity head on entering furnace, 1.0. Total 2.7.
- (11) Entry from air,  $f_1 = 2.0$ . Two 45° bends in butterfly valve at 0.5 each, 1.0; bend at bottom into flue, 1.5. Enlargement into flue, .05. Total 4.55.
- (12) Contraction from 15 square feet to 11 square feet,  $f_1 = 0.2$ . Division into three passages 2.0. Total 2.2.
- (13) Entry to uptakes 2.0. One 45° bend 0.5. Enlargement into port 0.8. Total 3.3.
- (14) Sharp bend into port 2.0. Exit to furnace 1.0. Total 3.0.
- (15) Entry from furnace 2.0. Enlargement to uptake 0.25. Total 2.25.
- (16) It is assumed that for the gradual widening of the gas uptake at the bottom the  $f_1$  value is negligible. The  $f_1$  value of 1.0 for the exit into the space above the chequers is therefore applied to the velocity leaving the widened portion of the uptake.
- (17) Enlargement on leaving passages below chequers:  $f_1 = 0.1$ . Although there is an apparent contraction from the free area of the chequer to the passages below, the velocity in these passages is gradually built up as the gases enter from the chequers over a considerable length. This is similar to the effect of a gradual contraction for which  $f_1$  is negligible.
- (18) The contraction in the outgoing flues is gradual, causing no  $f_1$  loss. Two rounded 45° bends gives an  $f_1$  value of 0.6 as in Note (6).
- (19) Contraction 8.75 square feet to 4.12 square feet, 0.89. Two 90° bends entering and leaving the valve at 1.5 each, 3.0. Valve itself, 2.5. Enlargement to 8.75 square feet of stack flue, 0.27. Total 6.46.
- (20) Entry.
- (21) Sharp bend and division into two passages, 2.0; contraction, 0.9. Exit at bottom of uptakes, 1.0; 45° bend, 0.5. Total 4.4.
- (22) Contraction from 8.75 square feet to 6.9 square feet, 0.21. Two 90° bends, 3.0. Valve itself, 2.5. Enlargement to stack flue, .05. Total, 5.76.
- (23) Length of stack flue between reversing valves—9 ft. approx. Perimeter of cross-section, 11.5 feet. Area of cross-section, 8.75 square feet. Hence  $f_2 = 0.15$ .
- (24) These velocities are calculated from the smaller area in the previous column, this being the area over most of the length of this particular section.

It will be understood that whilst the above example is intended to provide an illustration of how to deal with most of the bends, contractions and divisions usually met with in furnaces, each furnace must be treated on its merits. No general rules, except in the simplest cases, can be laid down, and choosing  $f_1$  for very complex bends and changes of section, often occurring together, is largely a matter of intelligent estimation until further experimental data become available. Whilst the total result must only be regarded as approximate, particularly as the internal lines of most high-temperature furnaces vary as the furnace gets older, calculation of the pressure balances by the above method is an invaluable guide as to whether bottlenecks exist, and to where alterations are required. Where quantities such as  $f_1$  are uncertain it is preferable to err on the high side as it is much easier to control flow in a generously designed furnace by dampers or chokes than to enlarge passages after they have been built too small.

TABLE 55. PRESSURE BALANCE FOR INGOING GAS

Calculation of  $p_v$ 

Part of furnace	Av. Temp. ( $^{\circ}$ C.)	Length $L$ (ft.)	Perim. $C$ (ft.)	Area $A$ sq. ft.	$v_o$ ft./sec. at N.T.P.	$P_o$ in. W.G.	$f_1$	$f_2$ from equn. (21)	$f_1 + f_2$	$P_o$ in. W.G.
Reversing valve .. ..	500	—	—	4.12	9.22	-0504	4.0 <sup>(6)</sup>	—	4.0	-2016
Flue .. ..	500	31	11.5	8.75-10.75	4.34 <sup>(24)</sup>	-0111	0.6 <sup>(6)</sup>	-51	1.1	-0123
Under chequers (3 flues) ..	500	—	—	7.5 <sup>(2)</sup>	5.07 <sup>(4)</sup>	-0152	2.3 <sup>(7)</sup>	—	2.3	-0350
Chequers (25 courses) ..	650	—	—	18.5 <sup>(2)</sup>	2.05	-0030	10.5 <sup>(3)</sup>	See Note 8	10.5	-0315
Uptakes (2) Lower mouths	800	8	5.33	5.6	6.79	-0379	2.0 <sup>(9)</sup>	—	2.0 <sup>(9)</sup>	-0768
Main portion ..				3.5 <sup>(2)</sup>	10.8	-0886	—	-30 <sup>(9)</sup>	.30	-0206
Ports (2) .. ..	800	3	3.6 <sup>(4)</sup>	1.8 <sup>(2)</sup>	21.6	-3650	2.7 <sup>(10)</sup>	.16	2.86	1.0439

TOTAL  $P_o$  (opposing flow) .. .. 1.4267

Calculation of  $p_s$   
 Buoyancy in chequers .. .. 12 ft. @  $650^{\circ}$  = .1286 in.  
 Net buoyancy in uptakes and ports .. .. 8 ft. @  $800^{\circ}$  = .0900 in.  
 Total buoyancy assisting flow .. .. .2186 in.  
 Less furnace pressure (opposing flow) .. .. .0100 in.  
 NET PRESSURE REQUIRED AT REVERSING VALVE 1.2181 in. W.G.

NOTE.—In Tables 55-57 and elsewhere in this calculation, four places of decimals have been used. Although the 3rd and 4th places of decimals have no practical significance they have been retained as making the calculations easier to follow than if the numbers had been rounded off to the nearest significant figure.

TABLE 56. PRESSURE BALANCE FOR INGOING AIR

*Calculation of  $p_v$* 

Part of furnace	Av. Temp. (° C.)	Length $L$ (ft.)	Perim. $C$ (ft.)	Area $A$ (sq. ft.)	$v_0$ ft./sec. at N.T.P.	$p_0$ in. W.G.	$f_1$	$f_2$ from equn. (21)	$f_1 + f_2$	$p_v$ in. W.G.
Reversing valve .. ..	50	—	—	6.9	5.64	-0090	4.55 <sup>(11)</sup>	—	4.55	-0410
Flue .. ..	100	17	11.5	8.75-15	4.45 <sup>(12)</sup>	-0065	0.6	.22	0.82	-0053
Under chequer .. ..	100	—	—	11	3.54	-0041	2.2 <sup>(12)</sup>	—	2.2	-0090
Chequers (25 courses) ..	500	—	—	22 <sup>(13)</sup>	1.77	-0021	10.5 <sup>(13)</sup>	See Note 8	10.5	-0221
Uptakes (2) .. ..	900	13	6 <sup>(1)</sup>	4.5 <sup>(14)</sup>	8.64	-0771	3.3 <sup>(13)</sup>	.35	3.65	-2814
Port .. ..	900	3	21	13.5	2.88	-0086	3.0 <sup>(14)</sup>	.04	3.04	-0281

TOTAL  $p_v$  (opposing flow) .. ..

-3849

*Calculation of  $p_s$* 

Buoyancy in chequers .. ..	..	..	..	..	12 ft. @ 500° = .1106 in.
Net buoyancy in uptakes and port ..	..	..	..	..	10 ft. 6 in. @ 900° = .1164 in.
Total buoyancy assisting flow ..	..	..	..	..	.2270 in.
Less furnace pressure (opposing flow) ..	..	..	..	..	.0100 in.

TOTAL  $p_s$  (assisting flow) .. ..

-2170

NET PRESSURE REQUIRED AT REVERSING VALVE .1679 in. W.G.

(See footnote to Table 55.)



**ILLUSTRATIVE EXAMPLES (other than for furnaces)**

In order to illustrate the use of the principles and equations here discussed some numerical examples are appended :—

(1) Clean producer gas (density at 60° F. and 30 inches bar is 0.07 lb. per cubic foot) is flowing through a 20-inch diameter main, in which is an orifice plate having  $\alpha = 0.61$  and 4 inches diameter. The pressure before the orifice plate 20 inches upstream is 8 inches w.g. and the differential pressure between this point and one 8 inches past the orifice is 2 inches w.g. If the barometer is at 29 inches and the gas temperature is 150° F., what is the quantity of gas passing per hour at 60° F. and 30 inches bar.

Solution :—

The conditions are similar to those of Fig. 59, and equation (12) can be used viz.,

$$v = \alpha \sqrt{2gh.}$$

The density of the producer gas at 150° F. and 29.6 inches bar. (i.e. atmosphere + 8 inches w.g.)

$$\times \frac{29.6}{30}$$

Since 1 inch w.g. = 5.2 lb./sq. ft., the height of a column of producer gas equivalent to 1 inch w.g. is

$$\frac{5.2}{0.07}$$

Thus,  $h$ , the differential head of 2 inches w.g. expressed as feet of gas

$$= 2 \times 88 = 176 \text{ feet}$$

$$v = 0.61 \sqrt{2 \times 32.2 \times 176} = 65 \text{ ft./sec. average velocity through the 4-inch orifice.}$$

This is now to be converted into cubic feet per hour,

$$\text{and} = 65 \times \pi \times \left(\frac{4}{12}\right)^2 \times 3,600 = 20,400 \text{ cu. ft./hr. at 29.6 inches and 150° F.,}$$

and at 30 inches and 60° F. this becomes

$$20,400 \times \frac{29.6}{30} \times \frac{460 + 60}{460 + 150} = 17,170 \text{ cu. ft./hr.}$$

(2) (a) A mineral oil of specific gravity 0.9 is to be pumped from a storage tank at ground level to discharge freely into an overhead tank 200 feet away and 30 feet above ground level. If the pipe is 3 inches i/d, and it has to take a maximum of 2,000 gallons of oil per hour, what pressure is required at the outlet of the pump at the storage tank (neglecting bends) ?

(b) A measuring device is installed in the main consisting of a 2-inch orifice plate in a 10-inch diameter chamber. A vertical U-tube containing mercury (sp. gr. 13.6) and oil shows 0.7 inch difference in levels. How much oil was passing at the time ?

(a) The flow of the oil through the pipes can be calculated by formula (5). Reynolds number must first be calculated.

$$Re = \rho \frac{Dv}{\eta} = \frac{Dv}{\nu}$$

$D = 0.25$  foot;  $\nu = 0.05 \times 0.0010764$  (Table 49) ; and since 1 cubic foot = 6.229 gallons—

$$\begin{aligned} v &= \frac{2,000}{6.229} \times \frac{1}{3,600} \times \frac{1}{0.125^2 \pi} \text{ ft./sec.} \\ &= 1.82 \text{ ft./sec.} \end{aligned}$$

## THE FLOW OF FLUIDS

$$\rho = 56 \text{ lb./cu. ft.}$$

$$\text{Re} = \frac{0.25 \times 1.82}{0.05 \times 0.0010764} = 3,453$$

whence, from Fig. 58,  $f = 0.011$ .

From equation (5)

$$\rho = \frac{0.25}{64.4} = 116.6 \text{ lb./sq. ft.}$$

To this must be added the head of oil in the 30 feet of vertical pipe, namely,  $56 \times 30 \text{ lb./sq. ft.} = 1,680 \text{ lb./sq. ft.}$

The total initial pressure is thus—

$$1,680 + 116.6 = 1,796.6 \text{ lb./sq. ft.} = 12.45 \text{ lb./sq. in.}$$

(b) The conditions are similar to Fig. 59 and equation (12) can be applied.

$$1 \text{ cubic foot of mercury weighs } 62.3 \times 13.6 \text{ lb.} = 847.3 \text{ lb.}$$

$$1 \text{ cubic foot of oil weighs } 56.0 \text{ lb.}$$

As the differential pressure is measured by a U-tube containing oil over mercury, the measured difference of 0.7 inch mercury in the level of the U-tube must be converted into feet of oil as follows :—

This correction means that the vertical height of 0.7 inch between the mercury columns is filled with oil and the weight of this oil (unlike the weight of a similar height of air) is appreciable and must be subtracted from the mercury gauge reading.

$$v = 0.61 \sqrt{2 \times 32.2 \times 0.824}$$

$$v = 4.45 \text{ ft./sec. through the orifice}$$

$$\text{Area of 2-inch orifice} = \pi/144 \text{ square feet.}$$

$$\begin{aligned} \text{Volume of oil passing} &= \frac{4.45 \times \pi \times 3,600}{144} \text{ cu. ft./hr.} \\ &= 349 \text{ cu. ft./hr.,} \end{aligned}$$

or, (since 1 cubic foot = 6.229 gallons) 2,200 gallons per hour.

(3) A Pitot tube is inserted in a 12-inch diameter gas main carrying blue water gas (sp. gr. 0.55, dry gas/dry air), along the centre line of the main. The differential pressure registered by a U-tube attached to the two connections and inclined to the horizontal at 10 : 1 is 0.5 inch w.g. The pressure in the gas main is 24 inches w.g., and the barometer pressure is 29 inches and the gas temperature 80° F. How much gas is the pipe carrying at 60° F. and 30 inches bar.?

$$\text{Total pressure in the main} = 29 + \frac{24}{13.6}$$

$$= 30.76 \text{ inches mercury gauge.}$$

The true pressure on a 10 : 1 inclination is 0.05 inch w.g.

The density of the saturated gas under the existing conditions of temperature and pressure can be calculated as follows :—

(a) Saturation pressure at 80° F. = 1.035 inch Hg.

(b) Volume of saturated gas at 80° F. and 30.76 inches Hg resulting from 1 cubic foot of dry gas at 32° F. and 29.92 inches Hg (N.T.P.)

$$= \frac{540}{492} \times \frac{29.92}{30.76 - 1.035} = 1.105 \text{ cu. ft.}$$

- (c) Under the above conditions, the weight of water resulting from 1 cubic foot of dry gas at N.T.P. will be :—

$$0.0502 \times \frac{1.035}{30.76 - 1.035} = 0.00175 \text{ lb.}$$

(0.0502 is the weight of water vapour per cubic foot at N.T.P.).

- (d) The density of the saturated gas under actual flow conditions will be as follows :—

Density of dry gas at N.T.P. =  $0.55 \times 0.0807 = 0.044385 \text{ lb./cu. ft.}$

Weight of water vapour =  $0.001750 \quad ,,$

Weight of 1.105 cubic foot of saturated gas =  $0.046135$

“ “ 1.0 “ “ “ “ =  $0.0418$

Head of gas corresponding to 0.05 inch w.g. =  $5.2 \times 0.05$   
=  $6.22 \text{ feet.}$

Equation (12a) may now be applied.

$$v = \sqrt{2 \times 32.2 \times 6.22} \\ = 19.6 \text{ ft./sec.}$$

This is the velocity at the centre of the main ; from Fig. 62,

$$\text{since : } \frac{\eta_{\text{max.}}}{\eta} \quad 0.0418 \times 1.0 \times 19.6 = 75,260$$

the ratio  $v_{\text{average}} : v_{\text{max.}} = 0.81$ ,

therefore average velocity =  $0.81 \times 19.6 \text{ ft./sec.}$

in terms of volume =  $0.81 \times 19.6 \times \pi \times 0.5^2 \times 3,600 \text{ cu. ft./hr.}$

=  $44,870 \text{ cu. ft./hr. at } 80^\circ \text{ F. and } 30.76 \text{ inches, sat.}$

=  $43,520 \text{ cu. ft./hr. at } 60^\circ \text{ F. and } 30 \text{ inches bar, sat.}$

- (4) What size of gas pipe is required to transmit 600 cubic feet of town gas (sp. gr. = 0.45) per hour to a point 100 feet from the gas meter where the pressure at the meter is 4 inches w.g. and the pressure required at the point of use is  $3\frac{1}{2}$  inches w.g. ?

If Lacey's formula (equation (8) ) is used, a reasonably probable value must be selected for  $f$  since  $Re$  cannot be calculated. When the size of pipe has been thus calculated, the value of  $Re$  can be calculated and the value of  $f$  ascertained more accurately ; it may then be necessary to repeat the calculation using this value of  $f$ .

It is more convenient for these low pressures to use Pole's formula (equation (9) )

$$600 = 1,350 \sqrt{0.45 \times 33.3}$$

when  $d = 1.43 \text{ inches diameter,}$

hence, a  $1\frac{1}{2}$  inch diameter pipe would be used.

- (5) Coke oven gas (sp. gr. 0.4) supplied by a booster giving a maximum pressure of 40 lb. per square inch is to be supplied over a distance of two miles through a main 10 inches diameter, the terminal pressure being 1 lb. per square inch gauge. What quantity of gas can be transmitted ?

It is necessary for the preliminary calculation to select an arbitrary value for  $f$ , say, 0.010.

$$0.010 \times 0.4 \times 10,560 \times \frac{10^5}{14.7^2} \times \frac{15.7^2}{14.7^2}$$

whence  $Q = 463,900$  cu. ft./hr. measured at  $60^\circ$  F. and 30 inches bar.

Under these circumstances, taking a section of the main where  $p = 3 \times 14.7$  lb./sq. in., the value of  $Re$  is calculated.

$$\rho = 0.0304 \text{ at S.T.P.} = 0.091 \text{ lb./cu.ft. in the gas main.}$$

$$D = 10/12 = 0.833 \text{ foot}$$

$$\eta = 1.37 \times 10^{-4} \times 0.0672 = 9.2 \times 10^{-6} \text{ f.p.s. units}$$

$$v = \frac{463,900}{3,600} \times \frac{14.7}{3 \times 14.7} \sim \frac{12^2}{\pi 5^3} =$$

$$\text{whence } Re = \frac{0.091 \times 0.833 \times 78.5}{9.2 \times 10^{-6}} = 646,800$$

whence (from Fig. 58),  $f = 0.0031$ .

Since this is markedly different from the assumed value of 0.010, it is necessary to recalculate the value of  $Q$  as above, using  $f = 0.003$ .

When this is done,  $Q = 847,000$  cu. ft./hr. at  $60^\circ$  F. and 30 inches bar.

Calculation of  $Re$  and reference to Fig. 58 shows that  $Re = 1,179,000$  and  $f = 0.003$ , whence the calculation is now found to be in order, since the correct value has been taken for  $f$ .



## CHAPTER X

## THE MEASUREMENT OF SOLID FUEL, STEAM, WATER, GAS AND AIR

Solid fuel; methods involving measurement and direct weighing—Measurement of fluid flow by orifice plate—Design data—Orifice location—Pressure losses—Formulae—Pulsating flow—Correction factors—Measurement of water by V-notch—Examples of calculation.

THE principles of the measurement of fluids have been discussed in Chapter IX, with special reference to the orifice plate, the Venturi meter and the Pitot tube. Equations (11), (12) and (12a) have been deduced as applying to these forms of measuring device. The practical application of the principles there set out is dealt with in this chapter. There are, of course, many commercial forms of measuring instrument, but these are not generally mentioned here, it being rather the object to describe basic methods.

Solid fuels come under a different category and accurate work necessitates some form of weighing machine.

## SOLID FUEL

Accurate determination of the quantity of fuel used is essential if the performance of a plant is to be known. The best method is to weigh the fuel on a weighing machine.

## INDIRECT MEASUREMENT

Where the coal allocated to individual plants cannot be weighed, a reasonably accurate figure can be obtained by volume measurement. If this is based on the volume of a coal hopper or bunker, it must be remembered that the volume weight may depend on the height from which it is dropped. Thus when a certain slack,  $0\frac{3}{8}$  inch in size and containing  $4\frac{1}{2}$  per cent. of moisture, was dropped from a height its volume weight was found to be as follows :—

Distance dropped	Wt. per cu. ft.—lb.
$1\frac{1}{2}$ feet ..	.. 47·57
2 „ ..	.. 47·61
4 „ ..	.. 49·00
6 „ ..	.. 50·48
8 „ ..	.. 51·48
10 „ ..	.. 52·35

The weight per cubic foot also depends on the size of the coal. Thus J. B. Deakin and W. T. K. Braunholtz ("Year Book," Coke Oven Managers' Association, 1930, 266), using a South Wales coking coal with 26 per cent. V.M. found :—

Size of coal	Wt. per cu. ft. lb.—lb.
2 -1 inch ..	.. 41·87
1 - $\frac{1}{2}$ „ ..	.. 42·78
$\frac{1}{2}$ - $\frac{3}{4}$ „ ..	.. 40·46
„ ..	.. 39·52

The dependence of volume weight on moisture content may also be considerable as is evident from Fig. 66, which refers to a slack coal. The absolute specific gravity of coals (i.e. specific gravity of the solid lump) may vary considerably and the volume weight will also depend upon the class of coal used. It will also depend on the way in which the coal "packs." A coal of mean sp. gr. 1·4 weighs some 87 lb. per cubic foot, whereas the nuts from such a coal from the figures given above weigh some 42 lb. per cubic foot. The air

spaces in the nuts thus occupy 52 per cent. If the coal consists of various sizes so that the fines can pack in the interstices between the larger pieces, the volume weight may increase considerably.

The volume weight of coke will depend not only upon its size, but upon the method of manufacture. The following figures may be quoted purely as an indication of the extent of these differences :—

	Bulk density—lb./cu. ft.	
	Size $\frac{3}{4}$ in.— $1\frac{1}{2}$ ins.	Rough
Metallurgical (oven) coke	.. 26–30	29–32
Horizontal retort coke ..	.. 23–26	25–28
Vertical retort coke ..	.. 21–22	23

These facts are given to indicate that when the quantity of coal or coke used as fuel is ascertained by volume measurements instead of by direct weighing,

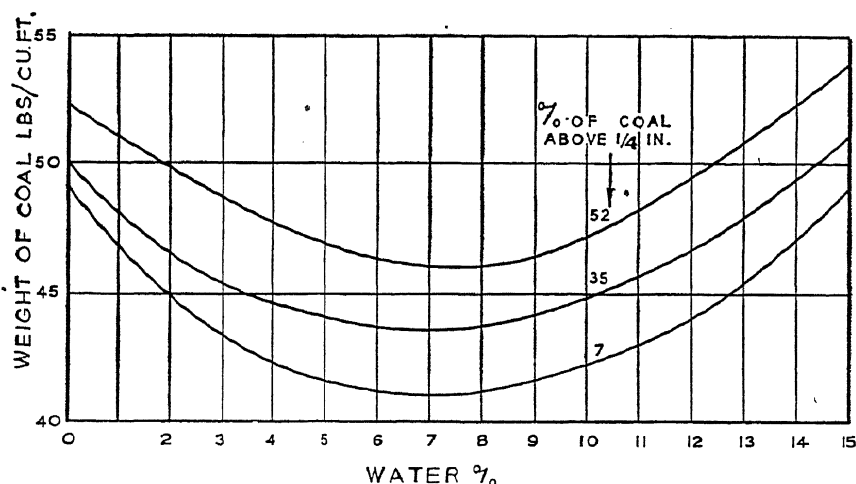


FIG. 66. Effect of water content on the volume weight of slack.

care must be taken to determine accurately the weight of fuel per cubic foot using the particular fuel supplied and the right conditions of packing.

One method of measuring fuel for small plants not provided with a weigh-bridge is to provide a gauge frame. This would not have a bottom, the floor serving instead. The frame is filled with fuel and levelled off by running a rod along the top edge, any surplus fuel being returned to the store. The frame should be of stout construction and provided with handles. If the amount of the fuel in use required to fill the frame is accurately weighed (preferably by more than one filling), a good indication of the total fuel used can be secured.

In small boiler houses, where the fuel is manually handled, a small weighing machine capable of weighing up to 4 cwt., could with advantage be installed. Most steel barrows used at these plants, either for moving coal or removing ashes, hold about 2 cwt. The weighing machine should be placed on a level part of the floor, in a convenient position. A small ramp made of heavy planks should be fitted at each side of the weighing machine platform to facilitate moving the loaded barrow over it. The arm of the weighing machine should be adjusted to balance exactly the weight of the barrow plus the weight of the fuel in it. Each time the full barrow passes over the platform the weight of fuel should be adjusted so that the weighing machine is balanced.

Whatever method of weighing or measuring is employed, it is desirable that the records of the weight of fuel used should be checked against the suppliers' delivery weights, preferably weekly. Variations in the quantity of fuel in stock should be taken into consideration when the comparisons are made.

### DIRECT WEIGHING

Direct weighing equipment takes many forms, and those most usually found are :—

- (a) A weighbridge placed in the railway tracks for weighing full coal waggons : where it is possible to weigh the same waggons, when empty, on the same weighbridge, the exact weight of fuel used can be ascertained.
- (b) Flush weighbridge for weighing fuel delivered in motor or horse-drawn lorries or carts. These machines give accurate weights and the previous remarks about re-weighing the empty vehicles apply.
- (c) Small portable weighing machines which can easily be erected on the stokehold floor. They are suitable for weighing fuel transported in small trucks or hand barrows.
- (d) Weighing machines, usually automatic, are sometimes placed in the stream of fuel, between the coal elevator and the bunkers or between the overhead bunkers and the chutes to mechanical stokers ; in smaller plants, coal meters could be fitted at the mechanical stokers.

All weighing devices should be periodically examined, kept in a proper state of repair and in correct adjustment. It is recommended that this work should be entrusted to the makers of the weighing machines.

### COAL MEASUREMENT

- (a) Coal measuring devices are often attached to automatic stokers, and the design of these varies with the type of stoker concerned. In some chain or travelling grate stokers, the mechanism is linked up with the mechanism for operating the grates, and records the volume of fuel passed.

With mechanical stokers in which some form of ram is used to feed the fuel on to the furnace grate, the equipment is arranged to record the quantity of fuel passed, from the number of strokes of the ram. Devices of this kind should be frequently calibrated, especially in plants where the size and type of fuel used often changes.

- (b) An appliance for measuring the flow of coal down a chute is sometimes used. This consists of an endless chain, one strand of which runs in the coal in the chute. The movement of the chain rotates a spindle which operates a counter from the readings of which the quantity of coal passed can be computed.
- (c) A measuring drum attached to the outlets from overhead bunkers which measures the quantity of coal flowing to a chute.

The drum is mounted on trunnions, and when full rotates, thus cutting off the flow of the coal to the bunker and discharging its contents into the chute. The number of times the drum is rotated is recorded by suitable mechanism, and if the weight of fuel the drums holds when full is known, the total weight of fuel passed can easily be checked.

### THE MEASUREMENT OF FLUID FLOW IN PIPES BY MEANS OF DIFFERENTIAL PRESSURE DEVICES

The metering of fluid flow in a pipe line by measuring the pressure drop caused by the insertion of a throttling device (Venturi tube, nozzle or orifice

plate) has proved to be one of the most reliable methods of determining the weight or volume of any fluid passing through a specified pipe line.

As indicated in Chapter IX, the method of measurement depends on the change of pressure energy caused by a fluid flowing through a reduced cross-sectional area in a pipe line.

The differential pressure devices in this chapter will be confined entirely to orifice plates. Information on Venturi tubes and nozzles can be obtained from the British Standard Specification No. 1042, which covers the theory as well as the practical application of all flow measuring devices by the differential pressure method.

The reduced area caused by the insertion of an orifice plate creates an increased velocity at this point with consequent loss of static pressure; this pressure loss when measured under specified conditions enables the ratio of flow to be determined.

## SYMBOLS

The symbols that will be used in dealing with the measurement of fluid flow are given in Table 58. In Table 59 are collected some of the fundamental data that will be used.

TABLE 58. SYMBOLS USED IN CHAPTER X

Symbol	Description	Units
B	Barometric pressure .. .. .	ins. mercury
C	Basic coefficient of discharge (excluding velocity of approach factor E) .. .. .	ratio
	Diameter of main (internal) .. .. .	inches
	$1$	
E	Velocity of approach factor $\sqrt{(1-m^2)}$ .. .. .	
H	Differential pressures, as read, inches mercury under water. (For ring balance meters multiply 1,273 in equation (3) and 1,018 in equation (7) by 1.039 .. .. .	in. Hg.
	Mercury .. .. .	—
	Absolute pressure .. .. .	in. Hg.
S	Static pressure .. .. .	in. Hg.
T	Absolute temperature ( $460 + t^{\circ}\text{F.}$ ) .. .. .	$^{\circ}\text{F. abs.}$
V	Rate of flow, at metering (working) conditions of temperature and pressure .. .. .	cu. ft./hr.
	Rate of flow .. .. .	lb./hr.
d	Diameter of orifice .. .. .	inches.
h	Effective differential pressure, inches water gauge (12.6 H, when H = inches mercury under water = 13.6 H with ring balance meters) .. .. .	in. W.G.
g	Acceleration due to gravity .. .. .	ft./sec. <sup>2</sup>
m	Orifice area = $\left(\frac{d}{D}\right)^2$ .. .. .	ratio
	Pipe area .. .. .	
p <sub>1</sub>	Absolute pressure, up-stream tapping .. .. .	lb./sq. in.
p <sub>2</sub>	Absolute pressure, down-stream tapping .. .. .	lb./sq. in.
s	Specific gravity (dry gas relative to dry air) .. .. .	ratio
t	Temperature (as measured) .. .. .	$^{\circ}\text{F.}$
v	Mean velocity of fluid .. .. .	ft./sec.
$\alpha$ (alpha)	Actual coefficient of flow, including E .. .. .	ratio
$\epsilon$ (epsilon)	Expansion correction factor .. .. .	ratio
$\eta$ (eta)	Absolute viscosity at working temperature .. .. .	—
$\rho$ (rho)	Density of fluid .. .. .	lb./cu. ft.
N.T.P.	Volume at normal temperature and pressure, 32 $^{\circ}$ F. and 29.92 in. Hg, dry .. .. .	
S.T.P.	Volume at standard conditions of temperature and pressure, 30 in. Hg and 60 $^{\circ}$ F. saturated .. .. .	

CARRIER RING FOR ORIFICE PLATE  
STEAM MEASUREMENT

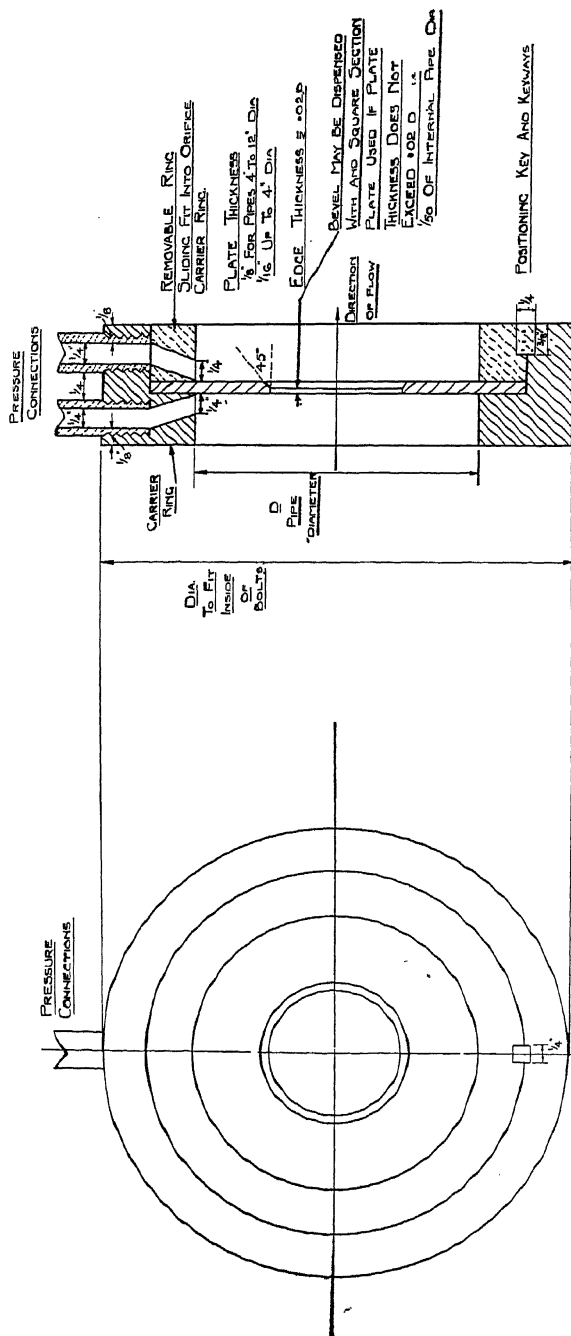


FIG. 67A. — Construction of an orifice plate.  
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TABLE 59. PHYSICAL CONSTANTS USED IN CHAPTER X

Density of water at 60° F.	= 62.35 lb./cu. ft.
Specific gravity of water vapour (dry air = 1)	= 0.624
g	= 32.2 ft./sec./sec.
1 cu. ft.	= 6.234 gallons.
Density of dry air at 30 in. of mercury and 60° F.	= 0.0764 lb./cu. ft.
Saturated water vapour pressure (w) at 60° F.	= 0.2563 lb./sq. in.
Absolute zero of temperature	= - 460° F.
1 in. Hg. pressure at 60° F. under air	= 0.490 lb./sq. in.
30 in. Hg. at 60° F. under air	= 14.70 lb./sq. in.
1 in. water gauge pressure at 60° F. under air	= 5.2 lb./sq. ft.
	= 0.03604 lb./sq. in.

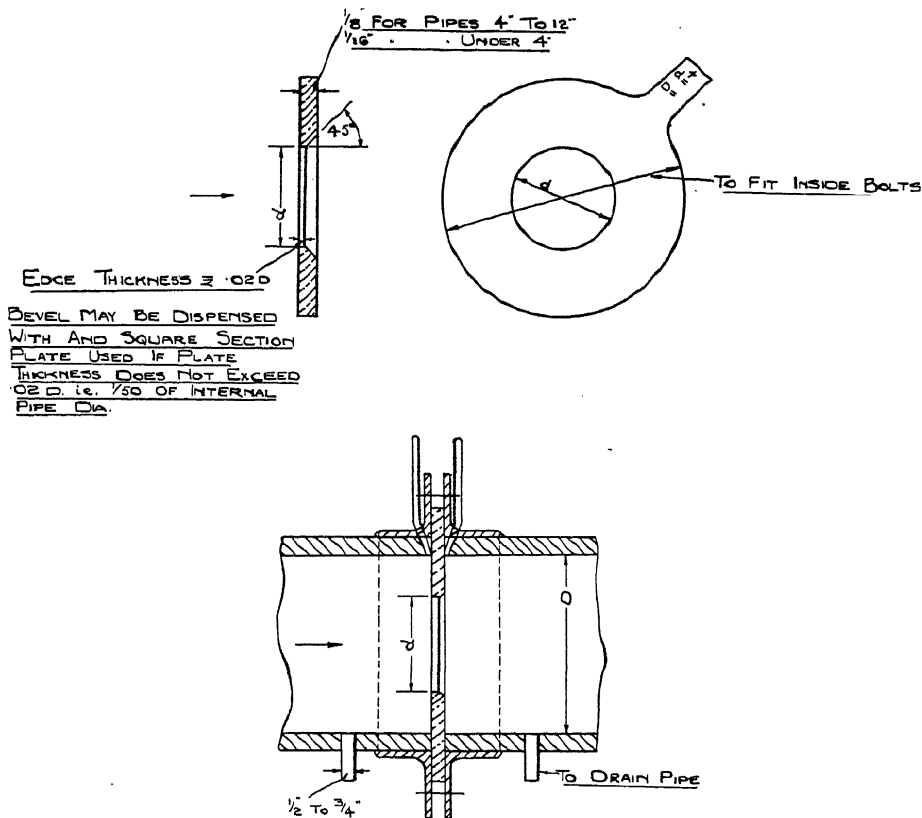


FIG. 67B. Construction of an orifice plate.

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## DESIGN DATA

(a) *General.* The basic design data covering the measurement of flow by orifice plates applies in general to all types of fluids.

The orifice plate is usually fitted between pipe flanges, and pressure connections are made on either side of the plate. An orifice may be used in any pipe not less than 2 inches internal diameter provided that "m," the ratio of orifice area to pipe area, does not exceed 0.7 and d, the orifice diameter, is not less than 0.25 inch. For pipe diameters less than 2 inches, it is preferable to order a

short length of pipe fitted with the pressure differential device as a complete unit from a meter manufacturing firm.

Typical orifice arrangements are given in Fig. 67 (a) and (b).

In Fig. 67 (a) the orifice plate is fitted to a carrier ring which also includes the pressure connections so that the whole may be inserted between pipe flanges at one operation. As an alternative for conditions when it is difficult to spring the flanges a sufficient distance to insert the carrier ring, method (b) may be adopted. In this arrangement the pressure holes should be drilled as close to the orifice plate as possible. This system of "corner tappings" as the position from which the pressure readings are taken is adhered to throughout this chapter.

The dimensions of the orifice plate and size and location of the pressure tappings must be adhered to when using the orifice discharge coefficients given later.

The orifice diameter ( $d$ ) must be calculated according to the quantity of flow, allowable differential pressure and the characteristics of the fluid flowing. It should be made to be correct to  $\pm 0.001d$ , and centred exactly in the pipe. The right angle edges on the square section plate and the right angle edge on the inlet side of a plate with a  $45^\circ$  level behind, must be sharp and free from ridges and not be rounded or broken.

The pressure connections should not project into the pipe; any projections or burrs should be filed off. It is equally important that jointing material does not protrude into the bore of the pipe. This can be avoided by cutting the jointing material with a hole  $\frac{1}{8}$  inch larger than the internal pipe diameter. Riveted flanges should also be avoided as the projecting rivets may introduce large errors in measurement. There should also be no change in pipe diameter in the immediate vicinity of the orifice. The position of the pressure tappings should be as specified in Fig. 67. If other positions are used a different discharge coefficient is required or a correction must be applied.

The size of the piping for connection between the meter and the orifice depends upon the distance the two are apart, but for normal lengths, say up to 60 feet,  $\frac{1}{4}$  inch I.D. pipe is satisfactory. For greater distance  $\frac{3}{8}$  inch or  $\frac{1}{2}$  inch I.D. pipe should be used.

(b) *Steam*. When measuring steam the following special precautions should be taken:—

(1) In horizontal pipes the pressure connections are best made on the side of the pipe that is brought out horizontally. With vertical pipes it is necessary to make the lower connection large, i.e.  $\frac{1}{2}$ -inch bore, and to bring it up to the level of the top connection. This is necessary to ensure that it is full of steam and not water. If it became filled with water a false head would be thrown on the meter. It is more convenient to measure on a horizontal pipe whenever possible.

(2) The pressure pipes should be full of water and this condition is obtained by installing condensing chambers at the point where the pressure tappings are made. The pipes transmitting the pressure from the orifice to the manometer or recorder must be free from leaks as a small leak (even though it might appear insignificant) may cause serious errors.

(3) There must be no air locks in the pressure pipes. Air is removed from the pressure pipes, when first connecting up to the manometer or recorder, by slacking off the unions at the instrument, blowing steam through to expel the air from the pipes and finally tightening up the unions.

(4) The condensing chambers can easily be made from standard pipe fittings; a simple type is illustrated in Fig. 68. The chambers should have sufficient area and condensing capacity to ensure that pressure pipes remain full of water and an equal head of water is thrown at all times on both pipes. The

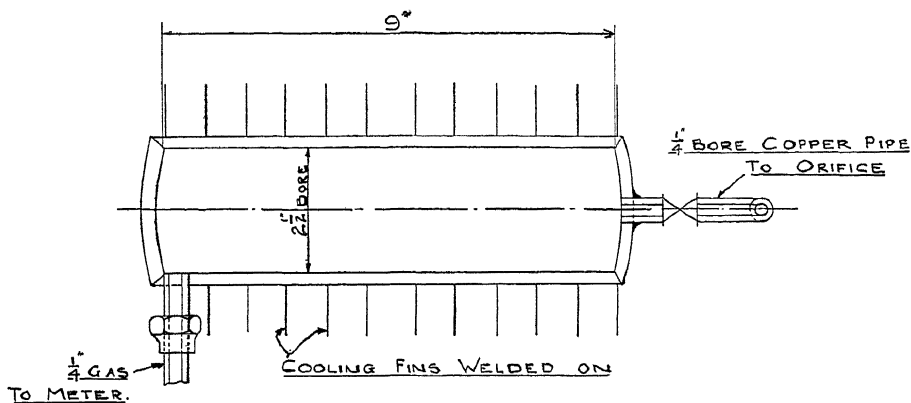


FIG. 68. Steam condensation chamber.

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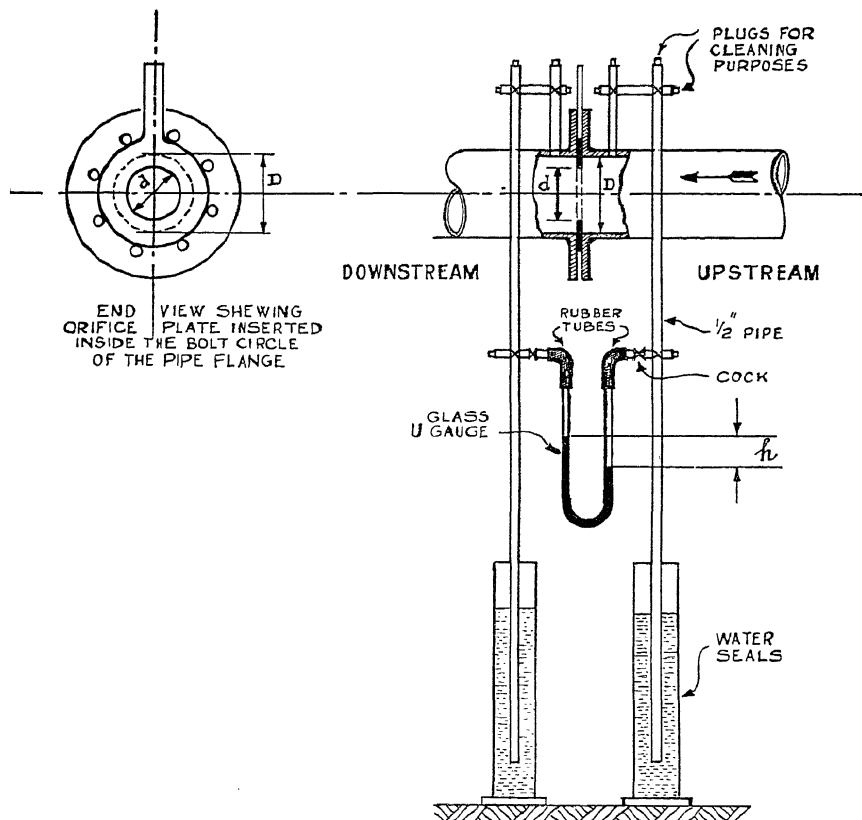


FIG. 69. Arrangement of orifice plate for gas measurement.

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design of the chambers should be such that the water displaced by the measuring instrument in moving from minimum to maximum reading or *vice versa*, does not affect the maintenance of an equal water head in both pipes.

(c) *Low Pressure Gases*. A simpler arrangement can be used when measuring the flow of gases and air at low pressure. The orifice plate used should be of the same design as that given in Fig. 67. On gas mains the size of the orifice should be arranged so that the differential pressure reading at maximum flow is reasonable, remembering that part of the pressure (cf. Chapter IX) is permanently lost. Fig. 69 indicates one arrangement of the apparatus. The U-gauge contains water. For more accurate readings an inclined gauge would be used. Whilst a very simple arrangement is sufficient for clean gas, it is necessary to prevent condensate from entering the U-tubes since this may alter the density of the liquid used and thus lead to uncertainty in the readings. Fig. 69 is a rather elaborate form of equipment designed with this object in view.

The straight length of pipe joining the orifice to the up-stream side should not be less than 10 pipe diameters long; on the down-stream side it may be less without the danger of introducing errors into the measurements. The off-takes must be arranged as close as possible to the orifice diaphragm. It is sometimes found desirable to enlarge the pipe diameter in the measuring stretch to accommodate the orifice plate, and when this is done the enlarged portion must conform to the lengths just given.

### ORIFICE LOCATION

The orifice should be placed in a straight length of pipe free from bends, valves, etc. The up-stream distance from the orifice plate must be adequate to eliminate the effect of any disturbance of the flow due to any of the causes described above. The same applies to the down-stream distance, but to a much lesser degree.

The minimum length of straight pipe preceding the orifice should be 8–10 pipe diameters and 3–5 diameters down-stream. Shorter lengths up-stream, especially when following bends and valves, etc., may involve an error of 2–3 per cent. For special cases, reference should be made to B.S.S. 1042.

Where the orifice edge towards the down-stream side is bevelled off at an angle of 45° care must be taken that the orifice is inserted correctly. With the bevel facing up-stream, a considerable error may be introduced, the readings being 20 per cent. too low.

### PERMANENT PRESSURE LOSS

In general the pressure differential should be made as large as possible to obtain greater accuracy on low rates of flow. The differential pressure that can be used is, however, generally limited by practical and economic considerations.

In determining the permissible maximum differential pressure, it is necessary to know the permanent pressure loss. This permanent loss varies according to the ratio of the orifice to the pipe diameter.

The percentage of the differential pressure which is permanent loss can be calculated with sufficient accuracy for practical purposes from the equation :—

$$\begin{aligned}\% \text{ loss} &= \left| 1 - \left( \frac{a}{D} \right)^5 \right| \times 100 \\ &= (1 - m) \times 100 \text{ where } m = \left( \frac{a}{D} \right)^5 \text{ (cf. Table 58).}\end{aligned}$$

Thus if  $m = 0.4$ , approximately 60 per cent. of the differential pressure is permanent loss.

## MEASURING INSTRUMENTS

The differential pressure may be measured by a manometer or recording meter according to the requirements. The type of instrument used depends upon the static pressure of the fluid and the range of differential pressure to be measured. When measuring the rate of flow of steam the manometer or "U" gauge must be capable of withstanding the maximum steam pressure.

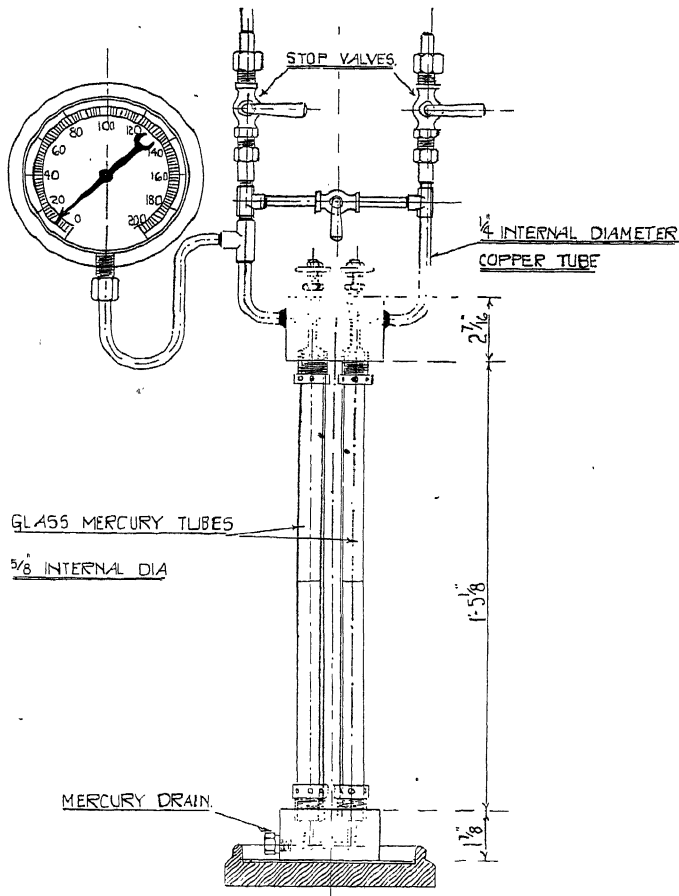


FIG. 70. Manometer for steam measurement.

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All metal parts liable to come into contact with the mercury should be made in steel. The tubes should be made sufficiently long to give adequate range to cover extreme conditions of flow. The manometer should be fitted with a valve to each limb and a third equalising valve on a connection below the other two. This enables the zero to be checked, and also safeguards the manometer from losing its indicating liquid should a sudden excess pressure occur on one side, when the limb valves are opened. Spindle screw valves should

preferably be used. A gauge to show the up-stream pressure should also be included. This is conveniently fixed to the high-pressure limb of the manometer. Such manometers may be purchased or made up in the workshop. A typical design is shown in Fig. 70.

For gaseous flow measurement, the differential pressure rarely exceeds 3 inches w.g., which allows the use of a "U" gauge of simple construction. For small differential pressures an inclined gauge can be used to give greater accuracy of reading.

Various types of recording instruments are available. They are usually calibrated in terms of rate of flow, and not in differential pressure. The scales can be graduated to indicate the flow at any specified standard condition for a given basic condition of flow, or on a decimal or percentage scale of a specified maximum flow.

### APPLICATION OF FORMULÆ TO THE MEASUREMENT OF STEAM, GAS AND AIR FLOW

The theory upon which the measurement of flow by a differential pressure device is based has been explained in Chapter IX. The differential pressure method is based on mass flow and the fundamental flow equation can be written as follows:—

$$W_s = \alpha \times \epsilon \times A \times \sqrt{2g \times \rho \times H_1} \quad . \quad . \quad . \quad (1)$$

$$V_s = \frac{W_s}{\rho}$$

$W_s$  = Weight of fluid flowing per second.

$\alpha$  = Coefficient of flow.

$\epsilon$  = Coefficient of expansion.

$A$  = Area of orifice in square feet.

$g$  = Acceleration due to gravity 32.2 ft./sec./sec.

$\rho$  = Density of fluid in lb./cu. ft. at conditions of flow.

$H_1$  = Differential pressure in lb./sq. ft.

$V_s$  = Volume of fluid flowing per second at existing conditions of temperature and pressure.

The coefficient of discharge  $\alpha$  depends on the ratio of the orifice area to the pipe area, expressed as  $m$ . Values for  $\alpha$  for a given value of  $m$  can be taken from Fig. 71. This flow coefficient  $\alpha$  is built up from a basic value  $C$  and a variable value  $E$ . For orifices with corner pressure taps, the value for  $C$  can be taken with sufficient accuracy to be 0.605. The value of  $E$ , called the velocity of approach factor, is determined by the formula given in Table 58.

The flow coefficient  $\alpha$  is  $C \times E$  and can, therefore, be determined for any size of orifice and pipe diameter.

There is one point, however, which must be borne in mind when designing an orifice plate for a given pipe size and for a definite flow. When the Reynolds number of the orifice (Chapter IX) is less than  $25 \times 10^4 \times d/D$ , the flow coefficient can no longer be considered to remain constant. Below that limiting value a correction must be applied or, if possible, the ratio of the orifice to the pipe area must be reduced.

In technical literature the Reynolds number is very often based on the diameter of the pipe, and not on the diameter of the orifice.

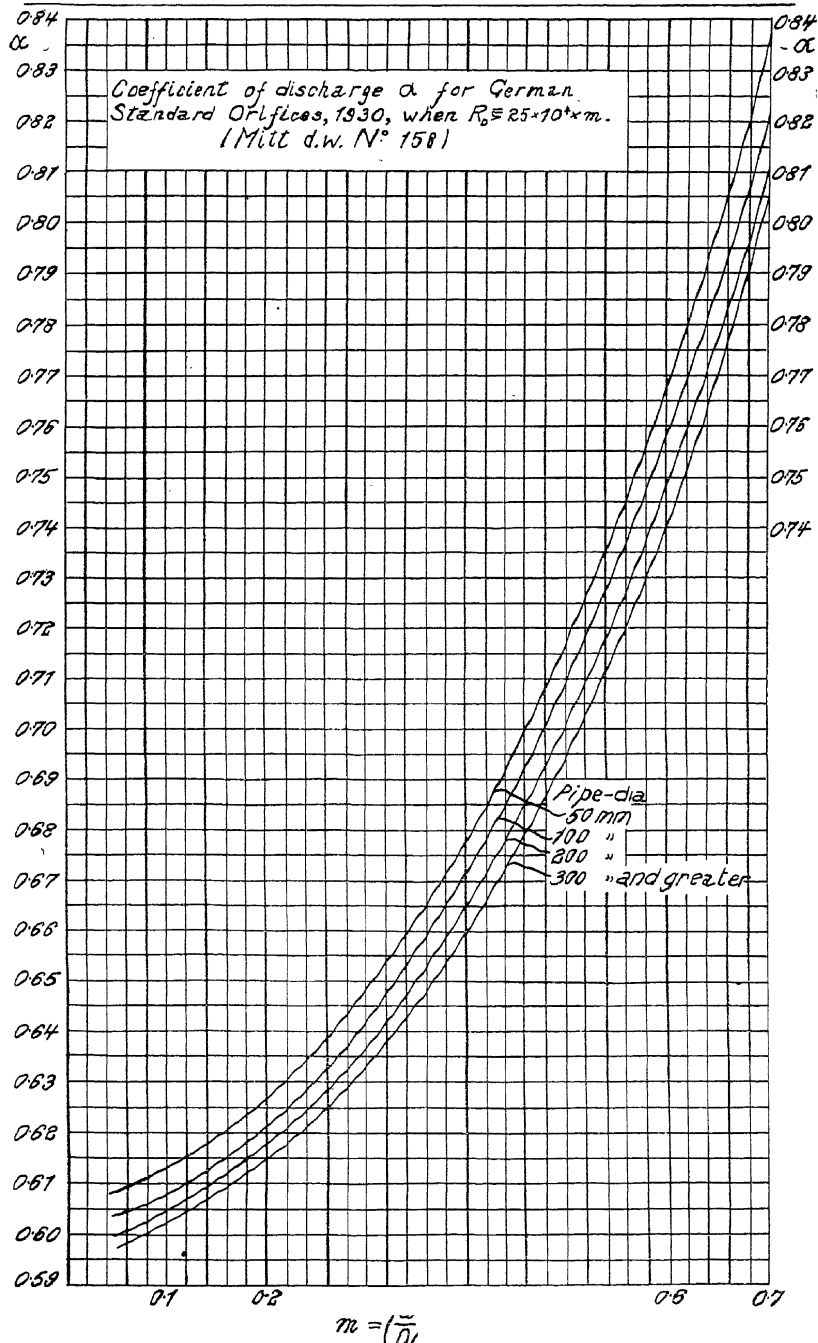
$$Re \text{ (orifice)} = Re \text{ (pipe)} \times D/d.$$

The conversion of the fundamental flow formula in self-consistent units into a formula for practical use, where the area is replaced by the diameter of the orifice, can be written as follows:—

$$W = 1,273 \alpha \quad \text{lb./hr.} \quad (3)$$

or

$$(4)$$

FIG. 71. Value of " $\alpha$ ."

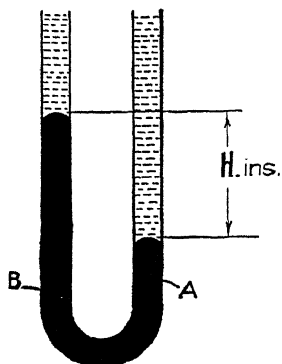
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For the determination of the volumetric flow, as in gas or air flow measurement, the formulæ can be expressed as follows:—

$$V = \frac{1,273 \alpha}{\sqrt{\rho}} \quad \text{cu. ft./hr.}$$

or  $\frac{\sqrt{\rho}}{\sqrt{\rho}} \quad \text{cu. ft./hr.} \quad (6)$

In formulæ (3)–(6),  $d$  is the diameter of the orifice in inches,  $h$  is the differential pressure in inches of water, and  $H$  is the differential pressure at orifice, as read, in inches mercury, under water. The expression “under water” means that the tubes above the mercury are filled with water. The additional weight of this water in the limb in which the mercury stands lower must be taken into account (Fig. 72).



Since mercury in limb A has an additional pressure of  $H$  ins. w.g. upon it above the water pressure in limb B, the true pressure showing on the gauge is:

$$H - \frac{H}{\text{sp. gr. of Hg.}} \text{ ins. Hg.}$$

or  $h = 12.6 \times H$  inches water gauge.

FIG. 72. Mercury gauge under water.

The orifice diameter required for given conditions of flow and differential pressure may be obtained by determining the value of  $x$  from the following equations:—

(a) Weight Units

$$\text{or } x = \frac{1,018 \epsilon W}{W} \quad (8)$$

(b) Volumetric Units

$$\text{or } x = \frac{287.4 \epsilon D^2 \sqrt{h}}{\sqrt{h}} \quad (9)$$

The corresponding value of  $d/D$  for a determined value of  $x$  can be read from Fig. 73 from which the required orifice diameter may be obtained.

The expansion correction factor  $\epsilon$  is introduced in the foregoing equations to

allow for the change in density between the two sides of the orifice. For normal conditions where the pressure drop across the orifice does not exceed 5 per cent. of the absolute pressure the correction can be omitted without introducing any serious error. The approximate errors introduced by omitting the correction are as follows :—

Differential pressure as percentage of absolute pressure,	'Error per cent. fast.
i.e. $\frac{P_1 - P_2}{P_1} \times 100$	
1.25 . . . . .	0.5
2.50 . . . . .	1.0
5.00 . . . . .	2.0

If greater accuracy is required, or if the pressure drop exceeds 5 per cent. of the absolute pressure, the correction factor can be obtained from the B.S.S. 1042.

For more accurate calculations the flow coefficient must be corrected for certain factors. These include :—

An extra tolerance for the Reynolds number, Re.

An extra tolerance for pipe or main diameter, D.

Complete information covering these points will be found in B.S.S. 1042.

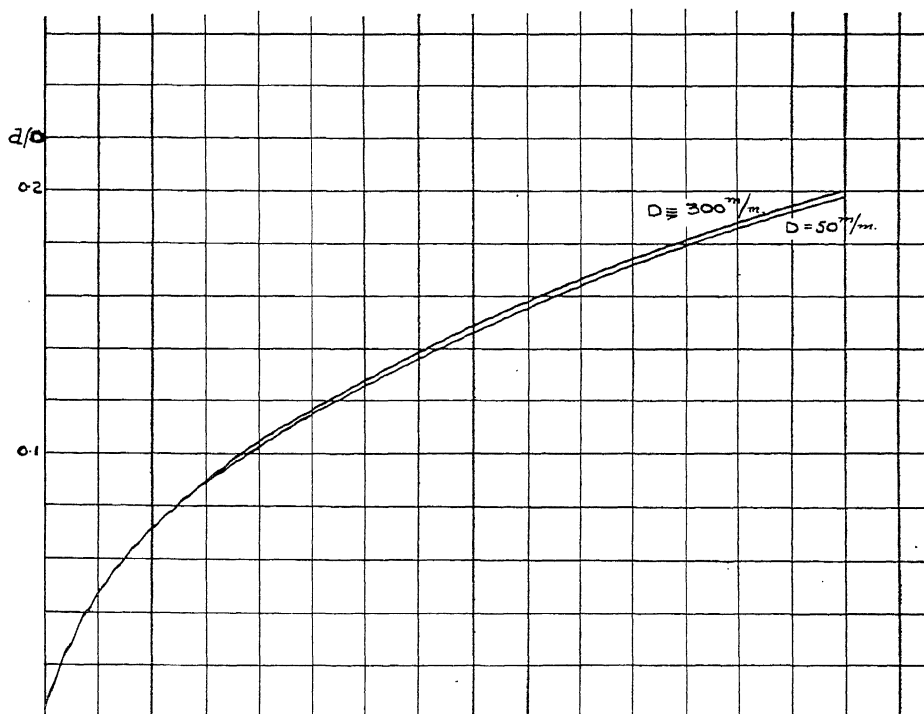


FIG. 73. Chart for determining orifice diameter.

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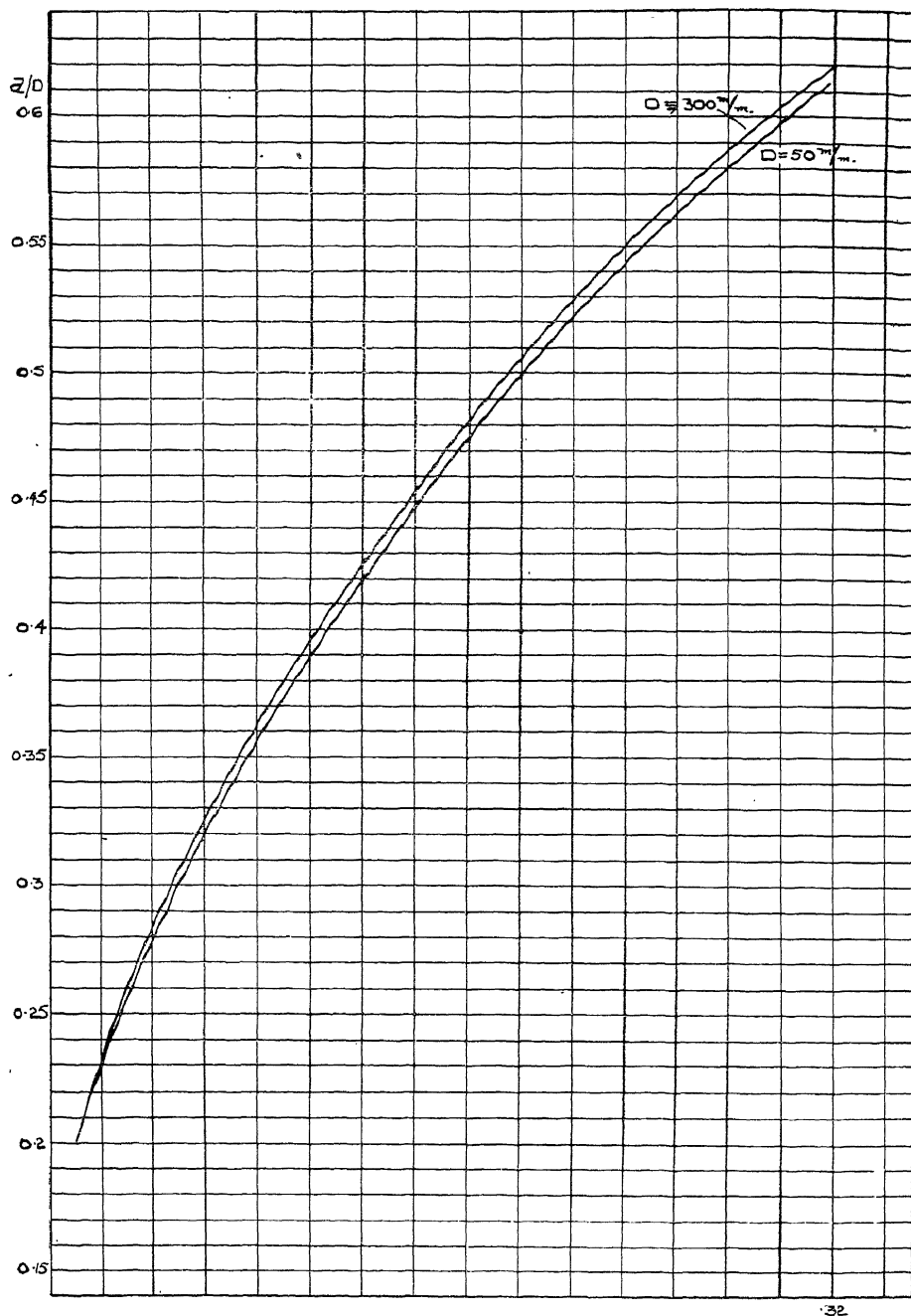


FIG. 73 (contd.)



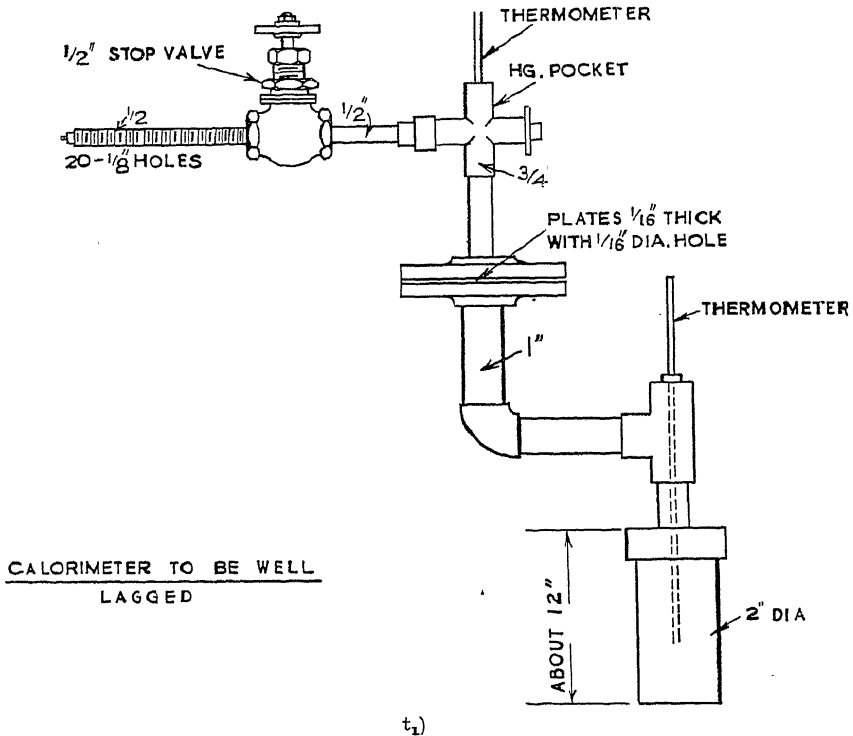
FIG. 7:



## MEASUREMENT OF DRYNESS OF STEAM

Steam meters are calibrated to work on specified conditions of the steam. When steam is not dry it is of great importance to the user to know the degree of wetness so as to apply the suitable correction to the steam meter readings.

One form of instrument which is commonly used and easy to construct is known as the "wiredrawing" or "throttling calorimeter" (Fig. 74). The action of this instrument depends on the fact that when wet steam at tem-



- where :  $Q$  = Quality of steam expressed as a decimal fraction.  
 $H$  = Total heat of dry saturated steam at atmospheric pressure.  
 $h$  = Heat of the water corresponding with the boiler pressure.  
 $t$  = Temperature of the steam after the orifice.  
 $t_1$  = Temperature of saturated steam at atmospheric pressure.  
 $L$  = Latent heat of steam at boiler pressure.  
 All values in B.Th.U.—lb.—°F units.

FIG. 74. Throttling calorimeter.

perature  $t_2$  and pressure  $p_2$  is allowed to escape through a small orifice, say  $\frac{1}{8}$  inch diameter, into another vessel at pressure  $p_1$  which is practically that of the atmosphere, droplets of water evaporate under the reduced pressure and dry steam is produced at a temperature  $t$ , which is higher than temperature  $t_1$  the temperature of saturated steam at pressure  $p_1$ . The dry steam is thus superheated. From a knowledge of the latent heat of steam at temperatures  $t_1$  and  $t$  and a measurement of the temperatures and pressures previously mentioned, the dryness fraction  $q$  can be calculated.

Both the high and low pressure chambers must be well lagged to prevent

heat loss or jacketed with steam at the same pressure and temperature. Steam should always be passed for a long enough period to establish a steady state before readings are taken. With these precautions, this instrument should be capable of considerable precision, but can only be used when the steam has a fairly high dryness fraction, e.g. when the wetness of the steam is below about 4 per cent.

The method of calculation can best be explained by a numerical example. Steam is obtained from a boiler operating at 100 lb. per square inch absolute and is expanded to atmospheric pressure, 14.7 lb. absolute. The temperature in the expansion chamber is 260° F. What is the dryness of the steam?

From steam tables,

	At 100 lb.	At 14.7 lb.
Temperature of steam—°F. . .	327.8	212
Sensible heat in water—B.Th.U./lb. . .	298.4	180.1
Latent heat of steam—B.Th.U./lb. . .	888.8	970.3
Dryness of steam—per cent. . .	x	Superheated to 260° F.

When the steam is expanded, the sensible heat in the water is reduced from 298.4 to 180.1, i.e. by 118.3 B.Th.U.; this heat is evolved.

The latent heat of the steam increases from 888.8 to 970.3 and this absorbs heat; the heat so absorbed is

$$\frac{x}{100} (970.3 - 888.8)$$

Any water present as such in the steam is converted into steam at 14.7 lb. and absorbs 970.3 B.Th.U./lb. of water, this being equal to

$$-\frac{x}{100} 970.3$$

The net result is that the heat evolved

$$= 118.3 - \frac{x}{100} (970.3 - 888.8) - \left(1 - \frac{x}{100}\right) 970.3 \text{ B.Th.U./lb. of total steam.}$$

This heat is given to the steam and superheats it. The degree of superheat is 260°–212° F., and since the sp. ht. of steam at atmospheric pressure is 0.46 B.Th.U./lb./°F. the heat measured as given to the steam is 0.46 (260–212).

Since heat evolved = heat given to steam

$$118.3 - \frac{x}{100} (970.3 - 888.8) - \left(1 - \frac{x}{100}\right) 970.3 = 0.46 (260 - 212)$$

$$\text{whence } 8.888x = (970.3 - 118.3) + 0.46 (260 - 212) \quad \dots (1)$$

$$\text{and } x = 98.33$$

From (1), by adding 180.1 on to 970.3 and 118.3, it will be evident that if

$Q$  = dryness fraction of the steam expressed as a decimal

$H$  = total heat of dry, saturated steam at atmospheric pressure

$h$  = sensible heat of water corresponding with boiler pressure

$t$  = temperature of steam after orifice

$t_1$  = temperature of saturated steam at atmospheric pressure

$L$  = latent heat of steam corresponding to boiler pressure

$$Q = \frac{(H - h) + 0.46 (t - t_1)}{L}$$

## PULSATING FLOW

The orifice method described provides an accurate method of measurement of what may be termed steady flows, that is, when the indicating gauge or meter is able to follow all changes of velocity. This, however, does not apply when the orifice is placed near a reciprocating engine, when serious errors in the meter readings may be introduced due to velocity pulsations.

Whenever a pulsating flow has to be measured it is advisable to consult the meter manufacturers.

### CORRECTION FACTORS FOR APPLICATION TO METER READINGS \*

A meter, unless fitted with an integral correction device, will give a true record of differential pressure only, and the conversion of the record into units of flow by a single multiplying constant can be carried out for one set of fluid conditions only.

For other conditions, the true rate of flow

= Flow obtained from meter readings at design conditions, multiplied by a correction factor F.

If F = Correction factor

P = Absolute pressure, up-stream tapping, in inches Hg

T = Absolute temperature of the gas in °F. at the same point as P

w = Vapour tension of water in inches Hg (taken from tables)

s = Specific gravity of gas (dry gas to dry air)

0.624 = Specific gravity of water vapour relative to dry air

b = Suffix to denote basic conditions for which meter was designed

m = Suffix to denote metering conditions,

the factor F for various units and fluids is determined as follows :—

(a) *Steam and liquids (lb. per hour).*

$$= \frac{V P}{T}$$

Correction factors for steam may be read from curves given in Figs. 75–78.

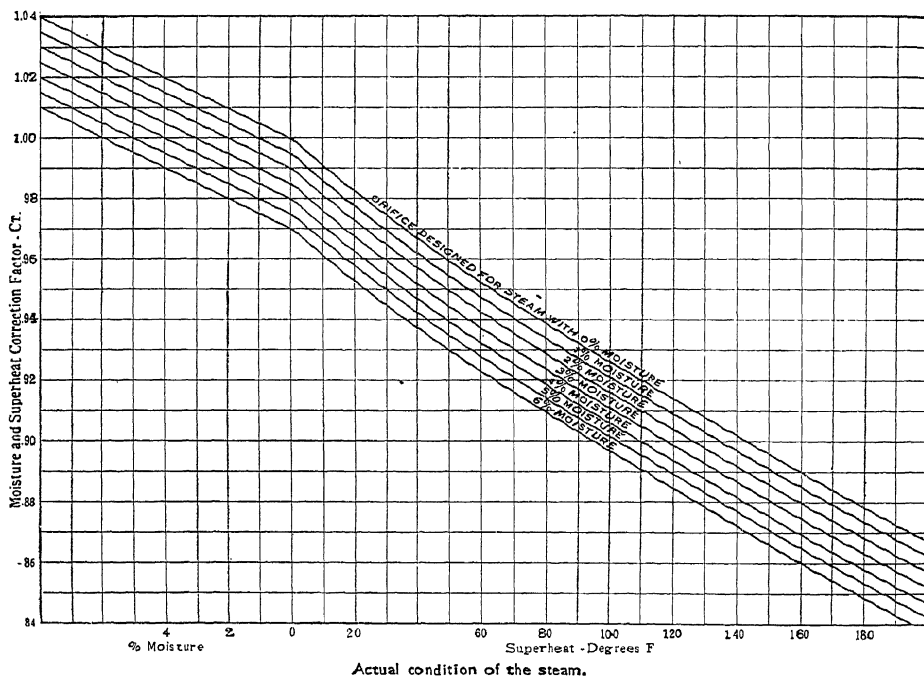


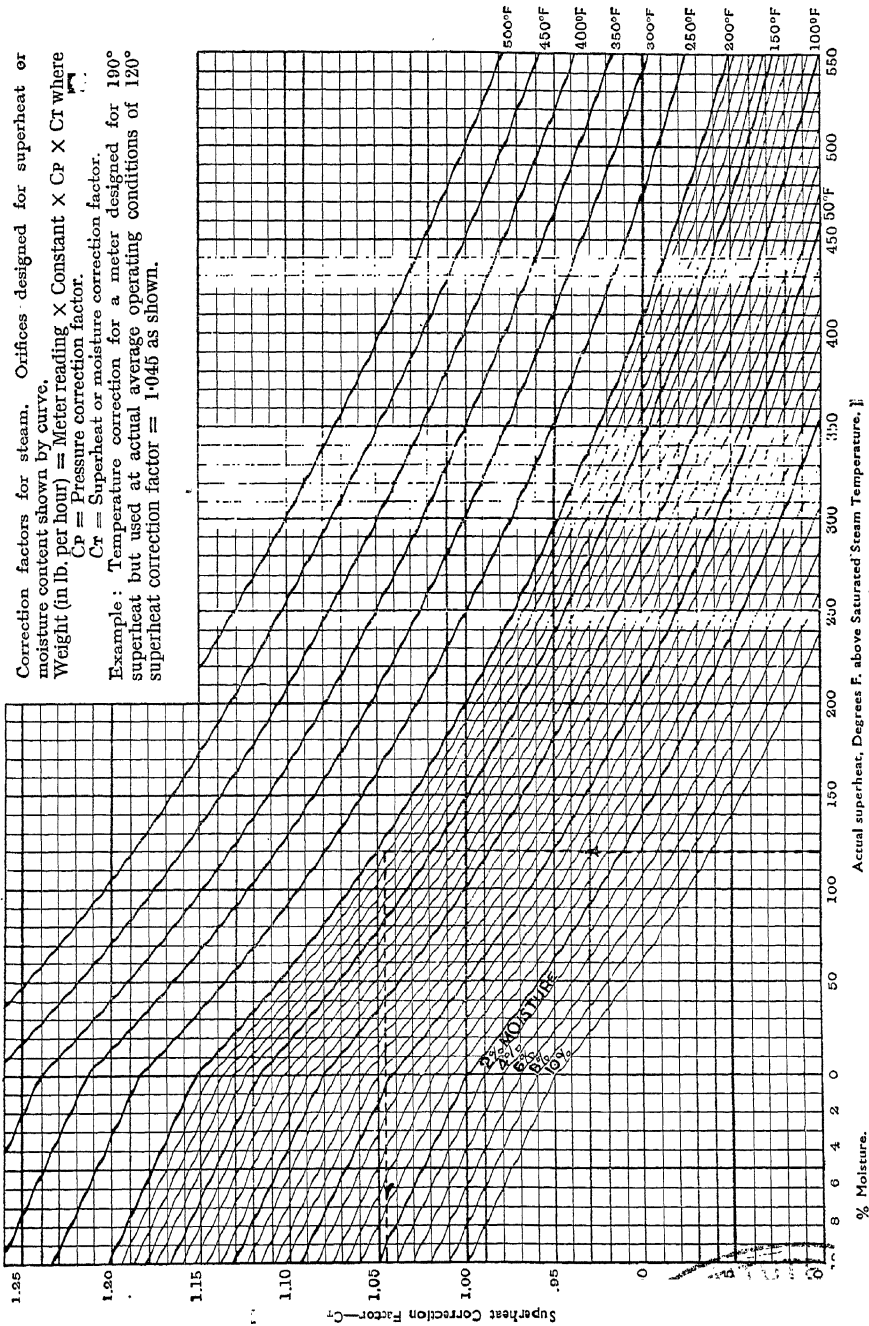
FIG. 75. Correction factors for steam meters.

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\* B.S.S. 1042.

Correction factors for steam. Orifices designed for superheat or moisture content shown by curve.  
 $\text{Weight (in lb. per hour)} = \text{Meter reading} \times \text{Constant} \times \text{Cp} \times \text{Cr}$  where  
 $\text{Cp} = \text{Pressure correction factor.}$   
 $\text{Cr} = \text{Superheat or moisture correction factor.}$

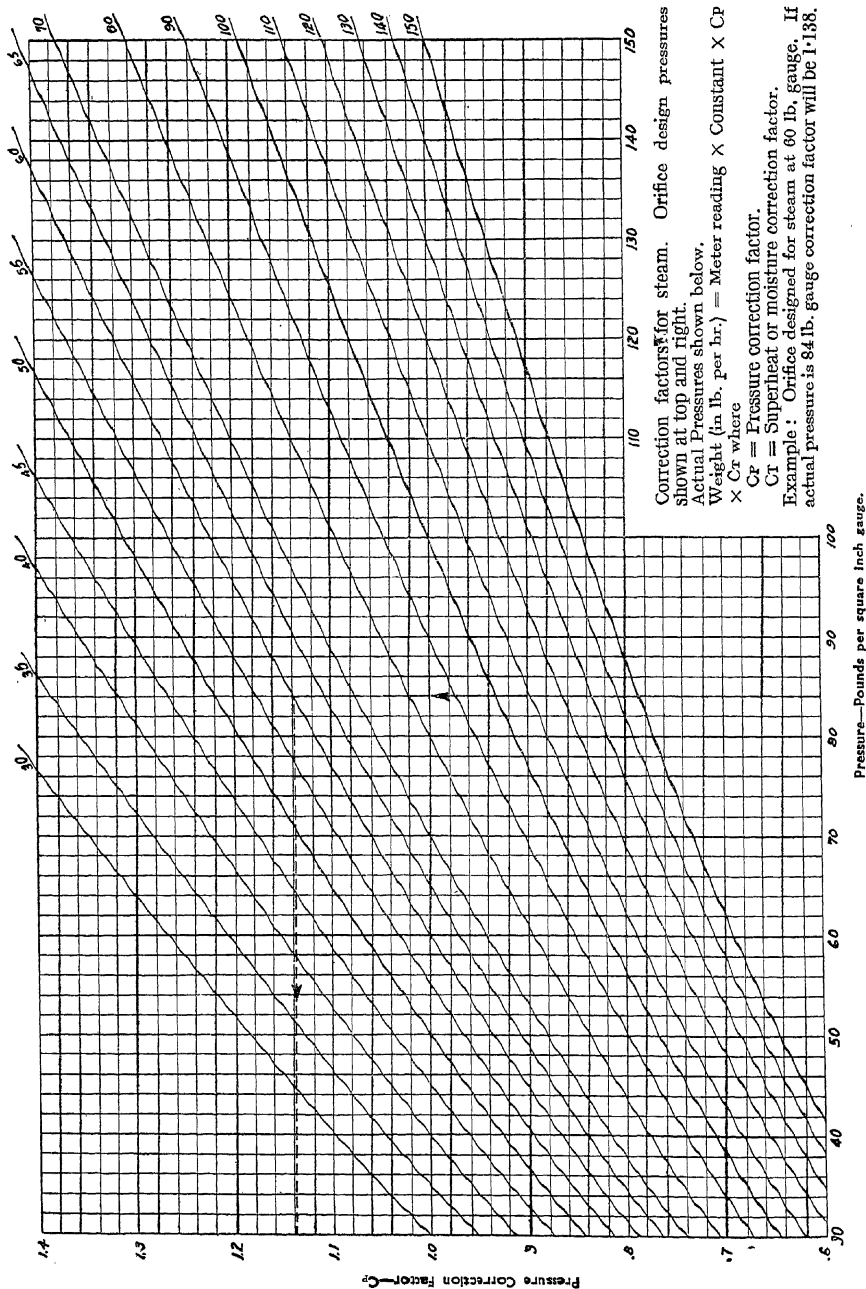
Example: Temperature correction for a meter designed for 190° superheat but used at actual average operating conditions of 120° superheat correction factor = 1.046 as shown.



Actual superheat, Degrees F. above Saturated Steam Temperature,  $t$

Fig. 76. Correction factors for steam meters.

(Reproduced from "Fuel Saving in the Iron and Steel Industry" by permission of the Iron and Steel Industrial Research Council.)



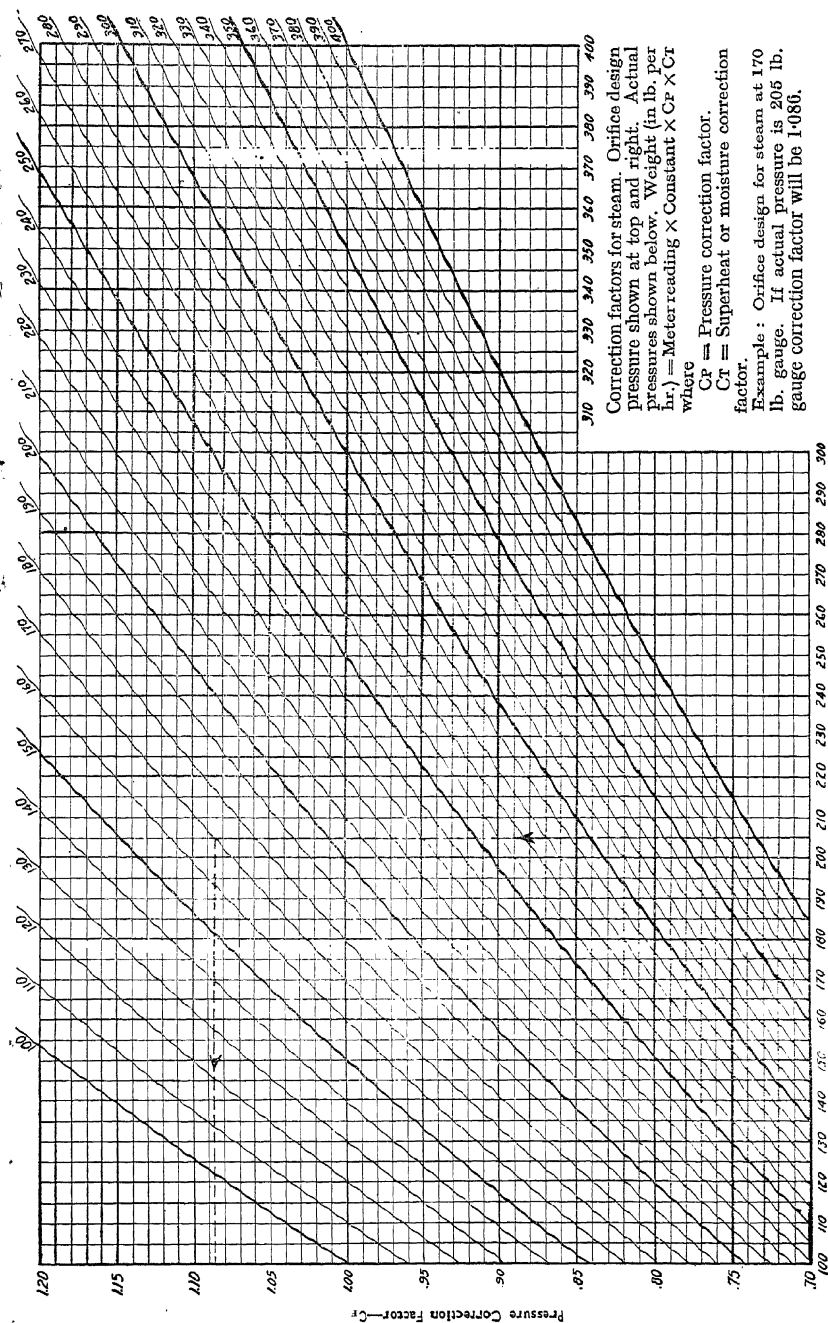


FIG. 78. Correction factors for steam meters.

(Reproduced from "Fuel Saving in the Iron and Steel Industry" by permission of the Iron and Steel Industrial Research Council.)

(b) *Dry air and dry gases.*

$$F = \sqrt{\frac{P_m \times \times}{P_b \times T_m \times s_m}}$$

(c) *Moist air and moist gas (saturated).*

$$F = \frac{0.624 w_b}{v - w_m + 0.624 w_m} \times$$

## THE MEASUREMENT OF WATER

The measurement of the water feed to the boiler is an important control of performance. Care must be taken, of course, that the water measured is evaporated in the boiler and that none of it is used for other purposes. There are many other occasions on which a measurement of water flow is needed. A device which can be easily constructed is that of a weir tank employing a V-notch. This will be described here. Other instruments that can be purchased for this purpose are described in Chapter XXIX.

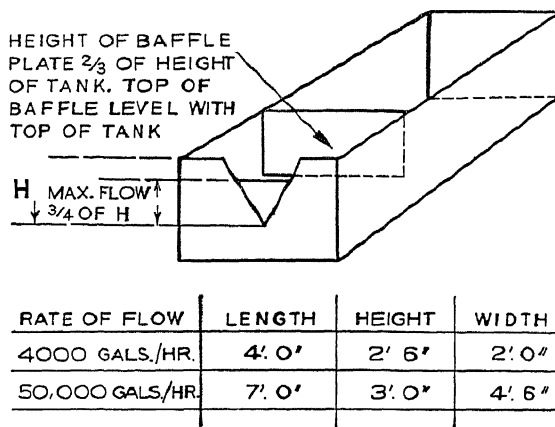


FIG. 79. Construction of a tank for V-notch measurement.

For the application of the weir tank (Fig. 79) the water must be at atmospheric pressure and below boiling point. The apparatus consists of a metal tank of a suitable size, fitted with a water-tight partition containing the weir, which may consist of a rectangular slot or a "V" notch (Fig. 79). The rate of flow is ascertained from the depth of water over the sill or apex of the notch. The rise and fall of the water can be measured by a float situated in a separate compartment on the up-stream side of the notch, this movement being transmitted to the indicator. Recording and integrating mechanism may be added if desired.

For accurate measurement, it is essential that the weir runs free, that is to say, the water flowing over the weir must have a free overfall. To ensure this, the level of the down-stream side of the notch must never be higher than say 1 inch below the sill or reversed apex of the weir.

The "V" notch proper should be made with sharp edges and of thin rustless metal plates. The slightest rounding of the up-stream edge increases the discharge appreciably. The inner face of the weir plate must be smooth, as any roughness or projection diminishes the upward velocity of the liquid, increasing the sectional area of the sheet of falling liquid and consequently the discharge also. The liquid must spring clear of the down-stream face, otherwise

the discharge is increased. The head ( $H$  or  $h$ ) is the distance (Figure in Table 60) between the crest of the notch or weir and the surface of the liquid at a point as near as possible to the outlet from the tank, but before the surface begins to curve downwards.

The tank may be fitted with an automatic regulating valve on the inlet side. This may be of the float-operated equilibrium type, with the float situated on the down-stream side of the notch. With this arrangement, if there is a tendency to drown the weir, the inlet valve automatically closes and reduces the rate of flow.

It will be readily appreciated that for accurate metering, steady flow conditions are required on the up-stream side of the notch; thus the measuring tank should not be used as a surge tank, and live steam should not be allowed to enter.

An installation of this description will always register the correct volume of flow irrespective of the temperature. It can, however, be calibrated to read the correct weight of flow at any desired temperature, and corrective factors are supplied by which the meter readings must be multiplied if the set temperature varies from that specified.

The general theory of the flow of a liquid over a weir, indicates that for the arrangement shown in Fig. 79,

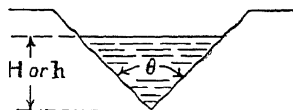
where  $Q$  is the quantity of liquid flowing in unit time and  $h$  the head.  $C$  is a constant depending on the density, viscosity and surface tension of the liquid, and on the angle between the sides of the notch (Table 60). For water, the density, viscosity and surface tension may be regarded as known—though they will vary somewhat with the temperature—and can be incorporated in the constant, leaving only the angle between the sides of the notch as the variable. For any given instrument this angle also is fixed when the arrangement is set up, and the flow can be read directly from a chart when  $h$  is measured.

Table 60 gives three formulæ and indicates their value, with the head ( $H$  or  $h$ ) as the only variable, for four selected values of the angle between the arms of the notch ( $\theta$ ). Table 61 gives for one of these formulæ complete details of the discharge under various conditions. Both these tables are taken from the handbook issued by George Kent Ltd.

TABLE 60. DISCHARGES OVER "V" NOTCHES

Symbols used :—

$H$  = height over notch in feet.  
 $h$  = height over notch in inches.  
 $Q$  = flow in cubic feet per second.  
 $G$  = flow in gallons per hour.  
 $\theta$  = angle of "V" notch.



Authority	General formulæ	Values of $\theta$			
		90°	60°	45°	30°
Approx.	$Q = 2.48 H^{2.5} \times \tan \theta/2$ $G = 111.5 h^{2.5} \times \tan \theta/2$	$2.48 H^{2.5}$ $111.5 h^{2.5}$	$1.43 H^{2.5}$ $64.4 h^{2.5}$	$1.03 H^{2.5}$ $46.2 h^{2.5}$	$.665 H^{2.5}$ $29.9 h^{2.5}$
Gourley & Crimp	$Q = 2.48 H^{2.47} \times \tan \theta/2$ $G = 120.1 h^{2.47} \times \tan \theta/2$	$2.48 H^{2.47}$ $120.1 h^{2.47}$	$1.432 H^{2.47}$ $69.34 h^{2.47}$	$1.027 H^{2.47}$ $49.74 h^{2.47}$	$.6645 H^{2.47}$ $32.17 h^{2.47}$
Barnes	$Q = 2.462 H^{2.48} \times Z$ $G = 116.3 h^{2.48} \times Z$	$2.48 H^{2.48}$ $117.1 h^{2.48}$	$1.435 H^{2.48}$ $67.8 h^{2.48}$	$1.032 H^{2.48}$ $48.73 h^{2.48}$	$0.669 H^{2.48}$ $31.61 h^{2.48}$

where  $Z = \tan \theta/2 / (\frac{1}{3} \sin \theta/2)^{.00703}$



TABLE 61. DISCHARGES OVER "V" NOTCHES  
(Calculated from Barnes' Formula)

Head in in.	Discharge in gallons/hour for angles of notch equal to :—										
	5°	10°	20°	30°	40°	45°	50°	60°	70°	80°	90°
2	29.10	58.03	116.4	176.4	239.1	271.9	305.9	378.2	447.9	548.8	653.6
4	162.3	323.7	649.3	983.7	1,334	1,517	1,706	2,110	2,449	3,061	3,646
6	443.8	884.9	1,775	2,689	3,646	4,146	4,664	5,768	6,830	8,369	9,967
8	905.7	1,806	3,622	5,489	7,442	8,462	9,520	11,770	13,940	17,080	20,340
10	1,575	3,141	6,300	9,547	12,940	14,720	16,560	20,480	24,240	29,710	35,380
12	2,476	4,937	9,902	15,005	20,340	23,180	26,020	32,180	38,110	46,690	55,600
14	3,628	7,236	14,510	21,990	29,810	33,900	38,140	47,170	55,850	68,430	81,500
16	5,053	10,070	20,210	30,620	41,520	47,210	53,110	65,680	77,770	95,290	113,500
18	6,787	13,490	27,070	41,010	55,600	63,250	71,130	87,960	104,100	127,600	152,000
20	11,080	17,520	35,150	53,250	72,210	82,110	92,370	114,200	135,300	166,700	197,400
24	18,810	27,540	55,240	83,710	113,500	129,000	145,200	179,500	212,600	260,500	310,100

### PRACTICAL APPLICATION OF THE FLOW FORMULÆ

The problems usually met in practical flow measurement with differential pressure devices, can be stated to be as follows :—

- (1) Determination of the rate of flow, for a specified orifice diameter and for a given or permissible differential pressure.
- (2) Determination of orifice diameter required to measure the flow for a specified plant. Here it is first necessary to make an approximation of the maximum quantity of the fluid to be measured and to decide on the allowable pressure loss in the pipe line.
- (3) For a specified maximum pressure loss (permanent loss), the  $d/D$  ratio must be carefully chosen, in order to be above the limiting value of  $Re$  at a considerably lesser flow. This applies especially to gaseous flow, where the available static pressure is very often less than 6 inches w.g. and where the permanent pressure loss must be kept at a minimum.

### PRACTICAL APPLICATION

The following examples will illustrate the practical application of the flow formulæ for various fluids.

#### EXAMPLE A. *Superheated Steam—Calculation of Flow*

Data obtained :—

Steam pressure  $p = 150$  lb./sq. in. gauge  
 $= 164.7$  lb./sq. in. absolute

Temperature of steam,  $t \dots \dots = 400^\circ \text{F.}$   
 Density,  $\rho \dots \dots = 0.340$  lb./cu. ft.  
 Differential pressure,  $H$  (as read)  $\dots \dots = 4$  inches Hg.  
 Pipe diameter,  $D \dots \dots = 8$  "  
 Orifice diameter,  $d \dots \dots = 4$  "

It is required to calculate the rate of flow in lb./hr.

For this example formula (3) is used.

$$W = 1,273 \alpha \epsilon d^2 \sqrt{H\rho} \text{ lb./hr.,}$$

but the correction for the expansion factor  $\epsilon$  is neglected.

Orifice ratio  $d/D = 0.5$ ,  $m = (d/D)^2 = 0.25$ .

Coefficient of flow  $\alpha$  (read from Fig. 71)  $= 0.627$ .

Rate of flow  $W = 1,283 \times 0.627 \times 4^2 \times \sqrt{4 \times 0.34} = 14,900$  lb./hr.

**EXAMPLE B. Saturated Steam—Determination of Orifice Diameter**

Data :—

Rate of flow, W (known or assumed)	= 16,000 lb./hr.
Absolute pressure, p .. ..	= 134.7 lb./sq. in.
Temperature, t .. ..	= 350° F.
Density, $\rho$ .. ..	= 0.3 lb./cu. ft.
Differential pressure, H (as read) ..	= 7.0 inches Hg
Expansion factor, $\epsilon$ .. ..	neglected
Pipe diameter, D .. ..	= 6 inches

It is required to determine what should be the diameter of the orifice, d.  
Here, formula (7) is used,

$$x = \frac{W}{1,018 D^2} = \frac{16,000}{1,018 \times 36 \times \sqrt{7 \times 0.3}} = 0.301.$$

Reading from Fig. 73, d/D for  $x = 0.301$  is found to be 0.605.

Since  $D = 6$  inches, the orifice diameter  $d = 6 \times 0.605 = 3.63$  inches.

When designing an orifice for use with a ring balance recorder equation (8) is used, the calculation being similar.

The Reynolds number for the foregoing example is determined as follows :—

$$Re =$$

$$\text{Here } D = \frac{6}{12} = \frac{3.63}{12} = 0.3025 \text{ feet}$$

$$v = \frac{16,000}{0.3 \times 3,600} \times \frac{144 \times 4}{3.63^2 \times 3.14} = 200 \text{ ft./sec.}$$

$\eta$  (poises) from Table 50 =  $160 \times 10^{-6}$  C.G.S. units.

Conversion factor into F.P.S. units from Chapter IX = 0.0672.

$\eta$  (British physical units) =  $160 \times 10^{-6} \times 0.0672 = 10.7 \times 10^{-6}$  F.P.S. units

$$Re = \frac{0.3025 \times 200 \times 0.3}{10.7 \times 10^{-6}} = 1,690,000$$

$$Re \text{ limit value } \dagger = 250,000 \times \frac{3.63}{6} = 151,000.$$

For the conditions specified above for temperature and pressure, the coefficient of flow will remain constant down to 10 per cent. of the maximum flow of 16,000 lb. per hour.

When the orifice has been designed for a given pressure and steam quality and actual conditions vary from this, correction factors to apply will be found from the curves in Figs. 75-78.

\* This is the nomenclature given in Chapter IX. For application here, "D" is the diameter of the orifice in feet, i.e. d/12.

† When the values of Reynolds number are sufficiently high the flow co-efficients remain substantially constant for standardised nozzles and orifice plates, i.e. they remain unaffected by the velocity, density and viscosity of the flowing fluid. The values of Re at which the flow coefficients can no longer be considered to remain constant are known as "limiting values" (B.S.S. 1042, p. 14).

**EXAMPLE C. Water—Calculation of Flow from Orifice Measurement**

Data :—

Diameter of main, D	.. ..	= 10 inches
Diameter of orifice, d	.. ..	= 7 inches
Temperature, t	.. ..	= 90° F.
Density at 90° F.	.. ..	= 62.11 lb./cu. ft.
Maximum differential pressure, H		= 6 inches Hg (as read)

It is required to determine the rate of flow, W, in lb. per hour and gallons per hour.

Formula (3) is used for this purpose.

$$W = 1,273 \alpha d^2 \sqrt{H\rho} \text{ lb./hr.}$$

Orifice ratio  $d/D = 0.7$ , whence  $m = 0.49$ .

Coefficient of flow  $\alpha$  (from Fig. 71) = 0.691.

$$\text{Rate of flow } W = 1,273 \times 0.691 \times 49 \times \sqrt{6 \times 62.11}$$

$$W = 831,880 \text{ lb./hr.}$$

Since 1 gallon of water at 90° F. temperature can be found from standard tables to weigh 9.961 lb.,

$$\frac{831,880}{9.961} = 83,500 \text{ gallons per hour.}$$

**EXAMPLE D. Coke Oven Gas (or Town Gas)**

Data :—

Maximum rate of flow, V	.. ..	= 250,000 cu. ft./hr. at S.T.P.
Maximum differential head, h	.. ..	= 2.0 inches w.g.
Diameter of main, D	.. ..	= 22 inches
Specific gravity (dry gas and dry air), s	.. ..	= 0.42
Density of gas, $\rho$	.. ..	= 0.0342 lb./cu. ft. at S.T.P.

It is required to determine what should be the diameter of orifice, d, for these conditions.

Formula (10) is used.

$$x = \frac{V\sqrt{\rho}}{\rho}$$

$$\text{neglecting } \epsilon, x = \frac{250,000 \times}{287.4 \times 484 \times \sqrt{2.0}} = 0.235$$

From Fig. 73 (b) for  $x = 0.235$ ,  $d/D = 0.545$ ,  
and therefore  $d = 0.545 \times 22 = 12$  inches.

An example of the calculation of the density of a gas is given in Chapter IX (pages 199–200).

**EXAMPLE E. Measurement of Water by "V" Notch**

Data :—

Angle of notch, $\theta$	.. ..	= 45°
Height over notch (inches)		= 6 inches.

It is required to ascertain the flow in lb./hr. when the water is at 140° F. From Table 61 the flow is 4,146 gallons per hour.

From standard tables the weight of 1 gallon of water at 140° F. is 9.8537 lb.

$$\begin{aligned} \text{Therefore, flow of water} &= 4,146 \times 9.8537 \\ &= 40,850 \text{ lb./hr.} \end{aligned}$$

## CHAPTER XI

## STOKING AND BOILER OPERATION

The air supply—The management of draught—Passage of air through a fuel bed—Methods and practice of hand firing—Methods and practice of mechanical firing—Water level—Blow-down—Cleaning fires—Banking fires—Cleaning heating surfaces—Carbon losses—Clinker formation.

**T**HE principles of combustion were discussed in detail in Chapters IV, V and VI. In the present chapter the application of these principles to practice is discussed with special reference to the burning of solid fuel. Stoking and boiler operation are here combined since most of the practical operation of the boiler is performed by the boiler fireman.

## THE AIR SUPPLY

Adequate draught and proper management of draught are the first requisites for proper combustion. The draught may be secured in one of the various ways mentioned in Chapters VI and XIV, but in whatever way it is obtained it is essential that proper use shall be made of it. The function of draught is to secure a sufficient flow of air through the furnace. It is the duty of the furnace operator to see that the air enters in the right place and flows through the right channels.

The principles of correct draught utilisation have been discussed in Chapter VI. It was there pointed out that between the inlet and outlet of a boiler or furnace system, including any auxiliary plant such as economisers and air heaters, there is a pressure difference which may amount to only a few tenths of an inch. This pressure difference is the motive power which moves the air and gases through the system. Flues and particularly auxiliary plant will absorb part of this pressure difference by their resistance. The firebed will also provide resistance. The importance of keeping the flues and other parts of the plant free from accumulations of dust, soot or other extraneous matter will be self-evident.

Air will flow by the path of least resistance and if leaks exist in the brickwork, air will flow through these rather than through the furnace. Leakage of cold air into the setting is one of the chief sources of fuel wastage, the waste arising from the lowering of the temperature of the gases with consequent reduction in the rate of heat transmission; it also reduces the available draught. The first step towards successful stoking is to make the setting air-tight. Methods of testing for leakage have been given at the end of Chapter VI. The objective should be to make the whole setting tight by putting it in thorough repair, and to do the work so well that it will stay tight. Nevertheless a thorough test should be made periodically, say once a month even under the best conditions.

The  $\text{CO}_2$  content of the gases, as indicated in Chapter IV, is the best general indication of the presence of undue quantities of excess air.

Leakages may occur in the following places :—

(1) *Economisers.* When scraper gear is fitted, a certain amount of air leaks through the holes through which this gear passes. Proprietary air excluders are on the market, but opinion seems to be somewhat divided as to their value.

(2) *External Brickwork.* Any expansion cracks or spaces in the brickwork, or between the brickwork and boiler shell, should be caulked with asbestos rope or pointed with fireclay mortar or caulked with a very thin mixture of fireclay and cotton waste, first pulling the waste apart so that the fibres will be covered with the clay.

In the smaller type of water-tube boiler, cracks may occur in the seating of

the drums or in the front doors. The brickwork is examined by holding against it an open flame as from a candle or oil lamp. Inleakage of air is indicated by the flame being drawn in. Places where this is found to occur are marked with chalk for attention.

The natural porosity of unglazed brickwork may be largely overcome by applying one or two coats of a mixture of hot tar and Portland cement of suitable consistency, or by coats of tar only, or by two or three coats of white-wash. The brickwork of the blow-down recess should be examined.

(3) *Dampers.* Sliding dampers often leak air, but there are various air excluders on the market which largely minimise this defect.

(4) *Ash Doors, etc.* Loss of effective draught may be caused through the door sealing the ash pit or the fire-door having become warped or burned. Damaged doors should be replaced as soon as possible. Furnace fronts, firing, inspection and cleaning doors should not be overlooked.

Leakages have a cumulative effect on draught because, first they increase the total volume of gases to be handled, and second they reduce the temperature of the gases and therefore reduce the chimney draught.

### MANAGEMENT OF DRAUGHT

The damper is the control valve of the combustion system, whether the draught be produced by a chimney or a fan. It is essential for proper control that :—

- (1) The damper shall be controllable from the firing floor so that the boiler attendant or furnace man does not have to walk some way (or even to climb over ladders, pipes, and so forth) to reach it.
- (2) The damper must be controllable inch by inch so that exact setting is possible.
- (3) A draught gauge should be so situated that when the damper is moved, the operator can see simultaneously the effect on the draught.

Where an induced draught fan is used, speed control should be adopted, thereby saving fan power, or alternatively the fan discharge damper should be used. All these methods lend themselves to some easily devised form of boiler front control. As an example, a boiler plant may be fitted with economiser swivel dampers provided with an arm on each spindle, the two linked by a flat iron link, a length of  $\frac{1}{2}$ -inch diameter wire rope passing over suitable pulleys to the boiler front and the dampers counter-balanced. The same arrangement can be applied to induced draught fans. Many existing boiler plants are not fitted with front damper control, but with a little ingenuity this can always be arranged. Without it, the damper is too often neglected.

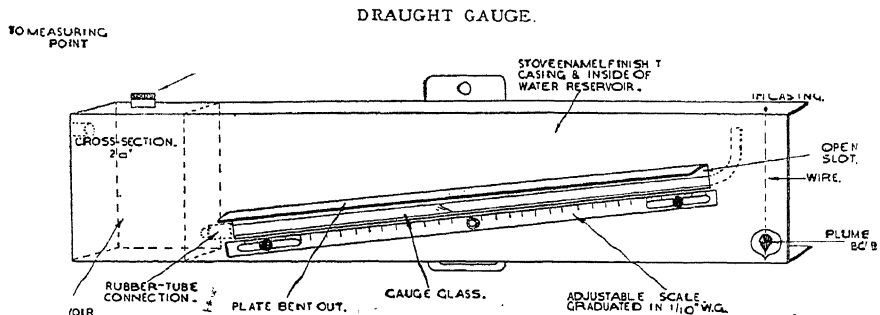


FIG. 80. Simple inclined draught gauge.

Quadrants with pin-and-hole adjustments are much too coarse for correct control. Where the main damper is the only control point and is of the usual rise-and-fall type and very heavy, a good plan is to install a light swivel damper on the boiler side and use this as a control.

The next point is to have means for measuring the draught. The extent to which the damper is closed is not an accurate measure of the draught, nor of the amount of air passing. Closing the damper by half will not reduce the draught to the same extent. Consequently a draught gauge is essential and the operatives must be taught how to use it. A simple draught gauge is shown in Fig. 80. Connections from all points at which draught is measured should be led to gauges placed near the firing floor so that the operatives can see the draught readily. The position of the gauges should also be arranged so that they can be easily read by the operator while adjusting the draught.

The correct draught for various conditions is a matter for trial, but control to less than 0.1 inch w.g. is necessary. If, for example, a draught at the chimney of 0.7 inch w.g. is correct, a draught of 0.8 inch may mean that the air supply is 12 per cent. greater than it should be.

Draught should be measured at the following points:—

- (1) Under the grate or other convenient point when forced draught is used.
- (2) In the boiler furnace or as near as convenient: for instance, in the Economic type of boiler the combustion chamber is a convenient position.
- (3) Either on the furnace side of the induced draught fan or at the base of the chimney, on the boiler side of the damper.

The correct draught can be established after seeing that all fires are in good condition, the fuel bed of the right depth for the type of fuel being burnt, and the grates well covered from back to front. The draught is then adjusted to be the minimum for the load required. When this point has been established it should be marked on the damper control. This figure will, of course, hold good only for the load operating at the time, and will require adjustment whenever the load varies. The air supply must be adjusted at all times to keep in step with the load.

Other methods of air control, such as closing the ashpit, are crude and ineffective and lead to other troubles, such as damage to firebars and clinker formation.

In furnaces equipped with forced draught, control of combustion will be effected by adjustment of the steam jets or of the damper controls provided, but it must be kept in mind that the chimney or induced draught must also be controlled so that the furnace is under either a very slight pressure or (for boiler furnaces) a very slight suction—in other words, in an almost balanced condition. Controlling the forced draught is only half the problem, and control of both forced and induced draught must go together.

The use of a draught gauge above the firing door (Fig. 81) indicating the draught in the furnace can be illustrated by assuming that a gauge is so connected as to register the draught in this position. If the firebed is allowed to burn too thin, or bare patches develop, the resistance of the firebed will decrease and the gauge reading will fall. If the fire is too thick, or becomes dirty, the gauge reading will increase, since the resistance of the fire has increased. Thus the gauge in this position will indicate the conditions in the firebed. A sensitive gauge reading to, say, 0.01 inch of water is desirable at this point.

The losses that may occur through wear of steam jets used for forced draught production has been discussed in the early part of Chapter VI. The steam jets should never be subjected to the full boiler pressure, and in general a reduction to about 30 lb. gives sufficient pressure to induce the necessary air.

Jets should be replaced when any appreciable wear is found to have occurred. It is advantageous to have the nipples made from stainless or non-corroding steel, phosphor-bronze or other metal which offers a high resistance to erosive wear. Jets must be correctly centred.

A valve should be provided to regulate the steam supply to the jets, and a

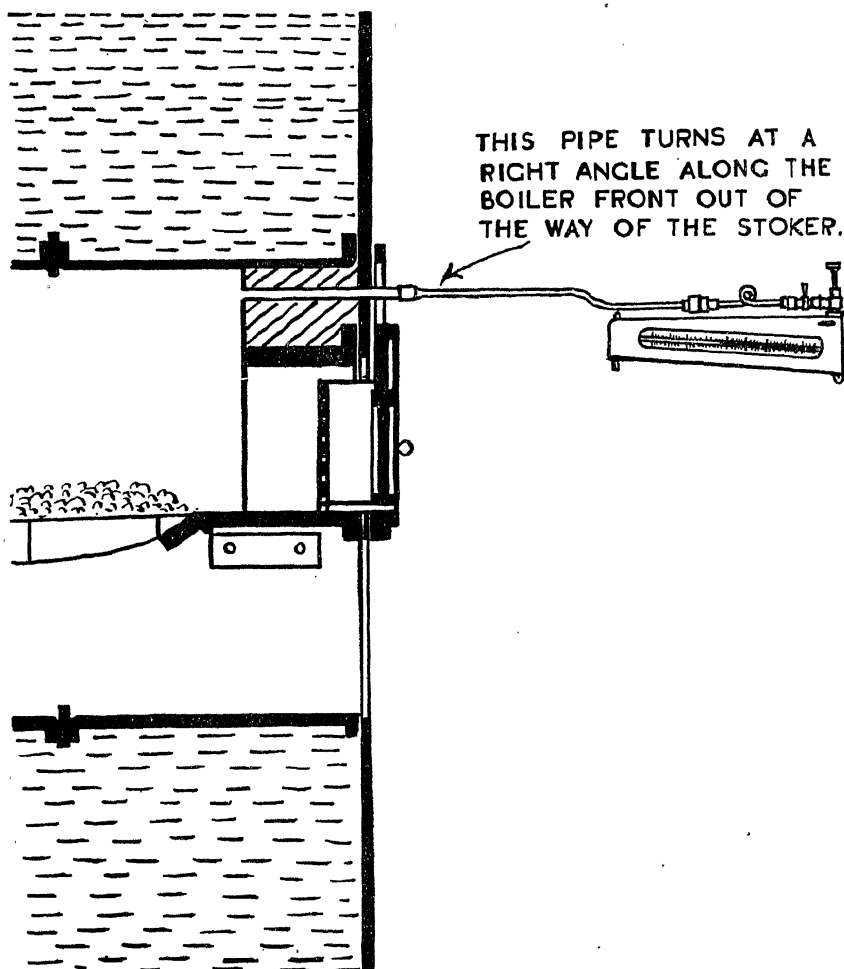


FIG. 81. Gauge in position on boiler front for taking draught over the fire.

pressure gauge fitted between the valve and the jets. The provision of a run of piping in the boiler flue is helpful in obtaining dry or superheated steam, thus improving the working of the jets.

Fans are discussed in Chapter XIV.

#### THE PASSAGE OF AIR THROUGH THE FUEL BED

When all the sources of loss due to friction and air leakage have been reduced to a minimum, the effective draught available will induce primary air through

the fuel bed and will draw in secondary air above the fuel bed. The air will take the path of least resistance, and if the rate of combustion in the furnace is not adequate the fault may lie in insufficient draught or motive power, or in faulty application of the draught.

If the grate is not covered by fuel at the back—which can be observed by looking under the bars—or if there are holes or thin places in the firebed the air will flow through these paths in preference to taking the path of greater resistance where the fuel bed is of the full thickness. The weight of coal burnt is decreased and the amount of excess air is increased when this happens.

As an example of the effect of maldistribution of the fuel the experience of a works may be quoted at which the steam output was insufficient apparently on account of shortage of draught. The conditions observed were as follows :—

Draught over fire	0·64 inch
CO <sub>2</sub> . . . . .	12 per cent.
Boiler output	10,220 lb. per hour

Correction of the fire conditions, consisting largely in securing an even and well-distributed fuel bed was followed by *reducing* the draught, with the following results :—

Draught over fire	0·32 inch
CO <sub>2</sub> . . . . .	14·5 per cent.
Boiler output	13,000 lb. per hour

It is a good rule to use as little draught as possible for the load to be carried.

Reference to Table 22 (Chapter V.) will indicate the serious effect of reduction in CO<sub>2</sub> content on the fuel consumption, an effect which becomes still more serious the higher the outlet temperature of the flue gases.

## RELATIONSHIP BETWEEN DRAUGHT AND RATE OF BURNING

The rate of combustion of a solid fuel will depend upon the rate at which air can be passed through the fuel bed. The size and coking properties of the coal as well as the thickness of the fuel bed will all have an effect upon this.

Some information on the draught required for various types of stokers is given later in this chapter.

## RATES OF COMBUSTION

The method of quoting rates of combustion in lb. per square foot of grate area per hour is misleading because of the widely differing calorific values of solid fuels and of the widely differing amounts of primary and secondary air required. A better principle would be to express the rate of combustion of solid fuel in therms per square foot per hour or B.Th.U. per square foot per hour. Since the weight basis is still in common use it is necessary to retain it here.

The limit to the economic rate is dictated by efficiency as reflected in the loss of carbon blown off the grate and up the chimney, loss of carbon in the ash, by maintenance problems connected with the very high temperatures developed at the high rates of burning, and by the use of power involved in the generation of draught.

## HAND FIRING

Three methods of hand firing are in use :—

- (1) Spreading or sprinkling method.
- (2) Side firing or wing firing method.
- (3) Coking method.



(1) *Spreading or Sprinkling Method.* This method consists of throwing an even layer of coal practically over the whole of the fire at each firing charge. Usually slightly less coal is placed on the back of the fire than upon the front, so as to prevent the production of smoke due to the cooling of the whole of the surface of the fire by the fresh coal. The back of the fire should never be bare, and it is necessary to use the rake to level up occasionally in order to make up for the lesser fuel charge upon the back portion of the furnace.

With care this method produces more steam than any other, and more uniformity, and should give a higher  $\text{CO}_2$ , but an unskilled fireman using it will produce more smoke. A good fireman is thoroughly master of the art of using his shovel and can place the coal exactly where he wishes on the grate. Strength is not the only requisite. The shovel should be used with a peculiar motion which only comes with practice. When the shovel is almost at the end of its throw, the extreme end of the handle should be smartly tilted downward, so as to "spray" the coal over a wider area. If the shovel is simply jerked from below the coal upon it, then the coal falls in a heap and gives the fire a very uneven appearance. This uneven thickness does not assist good combustion, and the rake has to be brought frequently into use to level the fire so as to allow the thicker portion to be thoroughly burned. A shovel of suitable size should be provided.

The spreading method is very useful for responding quickly to variations in steam demand, and for dealing with low-volatile fuels.

(2) *Side Firing or Wing Firing Method.* This method consists of throwing the coal first on the right-hand half of the fire and then allowing a period of time to elapse before the left half is fired. The left-hand side should not receive its charge until all the gases have been driven off from the first charge placed on the right-hand half. This method is widely used, and the advantages are that one half of the fire is always in good condition, and it ensures the combustion of the volatile matter given off by the other half. It helps to avoid black smoke.

*Rules for Hand Firing by the Spreading and Side Firing Methods.* Firing should be done lightly and often, the fire-door being kept open for the minimum time necessary. The firebars should be kept completely covered with coal, particular attention being paid to the back of the grate and the corners, right and left of the fire-door.

A level fire must be maintained, cracks and holes in it being filled with fresh charges of coal. The dampers are adjusted according to the load and secondary air admitted through the air-grids in the fire-doors as required. More secondary air is needed just after a fresh charge of coal than when the volatile matter has burned off. Excessive use of the poker, slice bar and rake must be avoided as it helps to cause clinker; moreover, excess air is admitted when the doors are open while tools are being used.

Dry slacks and finer coals are preferably wetted before burning. The wetting must be uniform and should be done some hours before the coal is used so that the moisture may become thoroughly distributed. If some parts of a heap are wet and others are dry, little advantage is gained by wetting.

(3) *Coking Method.* This method consists of charging the fresh coal on to the front part of the furnace to a depth of about 10 inches, and after allowing time for the gaseous matter to be driven off, pushing it forward over the rest of the grate. With this method very little smoke is produced as the volatile matter is burned when passing over the hot fire at the back, but it may not produce as much steam in a given period as either the spreading or side firing methods.

It is a useful method for small and lightly loaded boilers where the fireman has other duties to perform besides that of stoking.

*Thickness of Firebed.* The thickness of the firebed must be decided by the

skill and experience of the fireman according to the conditions prevailing. The size and nature of the coal are important; larger-sized fuels generally require thicker firebeds to prevent air passing through them unburnt.

The amount of fuel burnt, and, therefore, the amount of steam made, is determined by the quantity of useful air passed through a fire.

Too thin a fire leads to holes which cause (a) excess air, and (b) reduced output of steam.

Too thick a fire prevents air from passing through the fuel bed and so reduces the output.

For each coal there is a most suitable thickness of fuel bed which the fireman can discover by experiment. The way to do this is to fix a draught gauge above the furnace door in the manner previously described, and to determine by trials the relation between draught, fuel bed thickness and  $\text{CO}_2$  content.

It must be recognised that the thickness of bed with hand firing cannot be kept constant because the fire will be thickest immediately after firing and thinnest just before firing. It is, therefore, necessary to consider the mean thickness.

When hand firing it is difficult to get good results from a fire with a mean thickness of 3 inches or less owing to the ease with which air can pass through it, and the greater liability to form air holes. This liability will be increased, the higher the draught. Mean thicknesses of 4–6 inches have proved satisfactory. The thinner bed should be used with small coals such as  $\frac{1}{2}$ -inch slack or pearls, the bed should be thicker with  $1\frac{1}{2}$ -inch slacks or singles or nuts. It will also be found necessary to carry somewhat thicker beds when the ash content is high—say of the order of 15 per cent.

So long as the fuel bed is thin, little difference in performance will be found between the free-burning coals of groups 1 and 2, and the caking coals of groups 4 and 5. Thick beds will emphasise caking properties and cause the formation of crusts.

In plants where the load fluctuates considerably and there may be sudden demands for more steam, it may be advisable to carry slightly thicker fuel beds in order to have a greater reserve of hot coal on the grate.

*Use of Rake and Slice Bar.* The rake is used for levelling the fire. It should be used as little as possible as it tends to increase the danger of volatile matter escaping unburned and produce black smoke. It should be passed lightly over the fire surface. It should never be driven down on the firebars and pushed along as this tends to press the soft clinker between the bars, making it difficult to remove; it also mixes the ashes with the coal, and makes cleaning out longer and harder and more wasteful.

If the fire is heavy with ash, and air is not passing freely through it, the slice bar is run under the fuel and twisted slightly with a side movement. The ash must not be lifted into the upper zone and left there.

*Smoke Prevention.* To prevent smoke formation, the rich tarry gases evolved from the coal must pass over the incandescent firebed, and enough secondary air for combustion must be admitted through the openings in the fire-doors. This air must meet the volatile gases at a place where the temperature is sufficiently high for combustion to take place.

The coal should be fired in small amounts at short intervals and well distributed over the surface since in this way only small amounts of volatile matter are evolved per square foot of grate area at any one time. This does not apply to the coking method.

When a  $\text{CO}_2$  recorder is not available, a light smoke haze from the chimney is generally a good indication of satisfactory combustion conditions, provided the temperature of the fuel bed is high. When burning coke, anthracite or smokeless coals, the chimney is, of course, always clear.

*Cleaning Fires.* Cleaning takes several minutes, and before cleaning, the dampers should be checked as much as possible to avoid an inrush of cold air when the doors are open. Fires should not be burned down too low before cleaning, but sufficient fire must be left on the grate to start up rapidly after cleaning.

The water level should be so manipulated as to avoid the need for feeding water to the boiler while cleaning the fires, as otherwise the steam will be rapidly reduced. Times of cleaning should be arranged to fit in with periods of low load. Where there is a bank of boilers, fires should be cleaned in rotation.

For cleaning the fires the slice bar and hoe are used. The fires should be cleaned out thoroughly, one fire at a time, all ash and clinker being removed so that they will not fuse to the bars or bridge.

There are two methods of cleaning hand fired furnaces, namely, the side method and the front-to-back method. In the side method good coal is pushed from one side to the other, and the clinker and ash scraped out by the hoe after being loosened, if necessary, by the slice bar. The burning coal from the other side is then removed to the clean side. A few shovelfuls of fresh coal are added in order to have enough burning coal to cover the entire grate when the cleaning is done. The clinkers are then removed from the second half of the grate, after which the fire is spread evenly over the grate and built up gradually with fresh coal.

When starting to clean a thin fire it may be necessary to put fresh coal on the side to be cleaned last in order to have enough burning coal left to start a hot fire quickly.

In the front-to-back method, the burning coal is pushed with the hoe or rake against the bridge wall. It is usually preferable to clean one half of the grate at a time. The clinker is loosened with the slice bar and pulled out of the furnace with the rake and the burning coal spread evenly over the bare grate. If this method must be used while the load is on the boiler, the side method should be employed after the day's run is over so as to prevent a large accumulation of thick and hard clinker at the bridge wall.

Reasonable care should be taken to see that unburned fuel is not drawn together with the clinker and ash.

The clinker and ash should be drawn from the furnace on to an iron or steel barrow, wheeled outside the boiler house and slaked with water. On no account should the ashes and clinker be drawn on to the plates in front of the boiler and slaked there, as this practice leads to corrosion and wasting of the plates and if allowed against the boiler may in time ruin it.

Special care should be taken to see that no coke is discarded with the clinker and ashes. Coke recovered is put back on the fire.

The small coal that falls through the grate bars should be collected and burned with fresh coal.

*Dimensions of Grate.* Considerable economies can often be affected by using a grate of size suited to the load. There is some difference of opinion as to what should be the maximum length of the grate, some authorities maintaining that the 6-foot grate is too long for the best operating practice, and preferring a 5-foot grate.

It is found, however, that whatever be the initial length of the grate it should be used to the full capacity in order to provide a sufficiently hot fire for adequate radiation heat transmission in the combustion chamber. Economies have followed the practice of shortening the grate by building up the end near the bridge with refractory bricks when a reduced seasonal load requires less coal to be burnt for several weeks.

*Hand Firing with Slurry and other Low-grade Fuels.* The term "low-grade" fuel may be considered to refer to those products having a calorific value

of between 8,000 and 10,000 British Thermal Units per pound "as received," and includes belt pickings, middlings, duff, fines and slurry.

The behaviour of low-grade fuels varies very much in different types of grates and furnaces, but generally the best results will be obtained with furnaces of the forced draught type. Low-grade coals can usefully be blended with others of better quality.

With duffs, fines and slurries the fires must be raked at frequent intervals in order to keep the fuel bed open and to prevent the fires dying down. With inferior slurries or filter cake the firebed should be of good depth, say 10–12 inches to give the necessary "body" of heat to ignite the fresh raw fuel when fired. Good under-grate air pressure is necessary, not less than 3–4 inches water gauge.

*Outcrop Coals.* Considerable quantities of outcrop coals are coming on to the market. They vary from low grade to quite good fuels. No hard and fast rules can accordingly be laid down for their treatment.

## MECHANICAL FIRING

Mechanical stokers may be grouped under the following heads:—

- I. Stokers generally used for shell boilers (including vertical boilers).
- II. Stokers generally used for water-tube boilers.

It is only possible to give general information regarding the use and characteristics of mechanical stokers.

*I. Stokers for Shell Boilers.* There are three generic types: the Sprinkler stoker; the Coking stoker; and the Underfeed stoker.

(1) *Sprinkler Stokers.* These throw the fuel all over the grate by means of shovels, flippers, or rotary distributors (cf. Figs. 27 and 82). They imitate the spreading method of hand firing previously described. Sprinkler stokers may be, again, subdivided into two classes:—

- (a) When the coal is spread by a shovel working intermittently, the grate bars generally move, i.e. the grates are self-cleaning. The coal is thrown to different sections of the grate in turn by means of the differential lengths of the cams of the mechanism; this type also imitates in some measure the side firing system of hand firing (Fig. 83).
- (b) When the coal is spread continuously by a rotary distributor the grate is generally stationary. In this system the coal is spread over the whole area of the grate evenly and continuously (Fig. 84).

(a) *Shovel Sprinkler.* The thickness of the fuel bed should be approximately as follows:—

	Inches
Small slacks ( $\frac{1}{2}$ -inch) and small sized coals (pearls) of relatively low ash content, say under 10 per cent.	3–4
Larger slacks ( $1\frac{1}{2}$ inches) and larger sized coals (nuts) of relatively low ash content, say under 10 per cent.	.
Coals higher than 10 per cent. ash content	slightly thicker.

The shovels and springs should be inspected at regular intervals and properly adjusted.

(b) *Rotary Sprinklers.* The thickness of the fuel bed should be approximately 2–3 inches with slacks, rising to 4 inches with larger coal.

*Sprinkler Stokers in General.* It will be noted that the suggested fire thicknesses are less than those used for hand firing, and consequently it is necessary to watch the fire carefully to guard against uncovered or unevenly covered

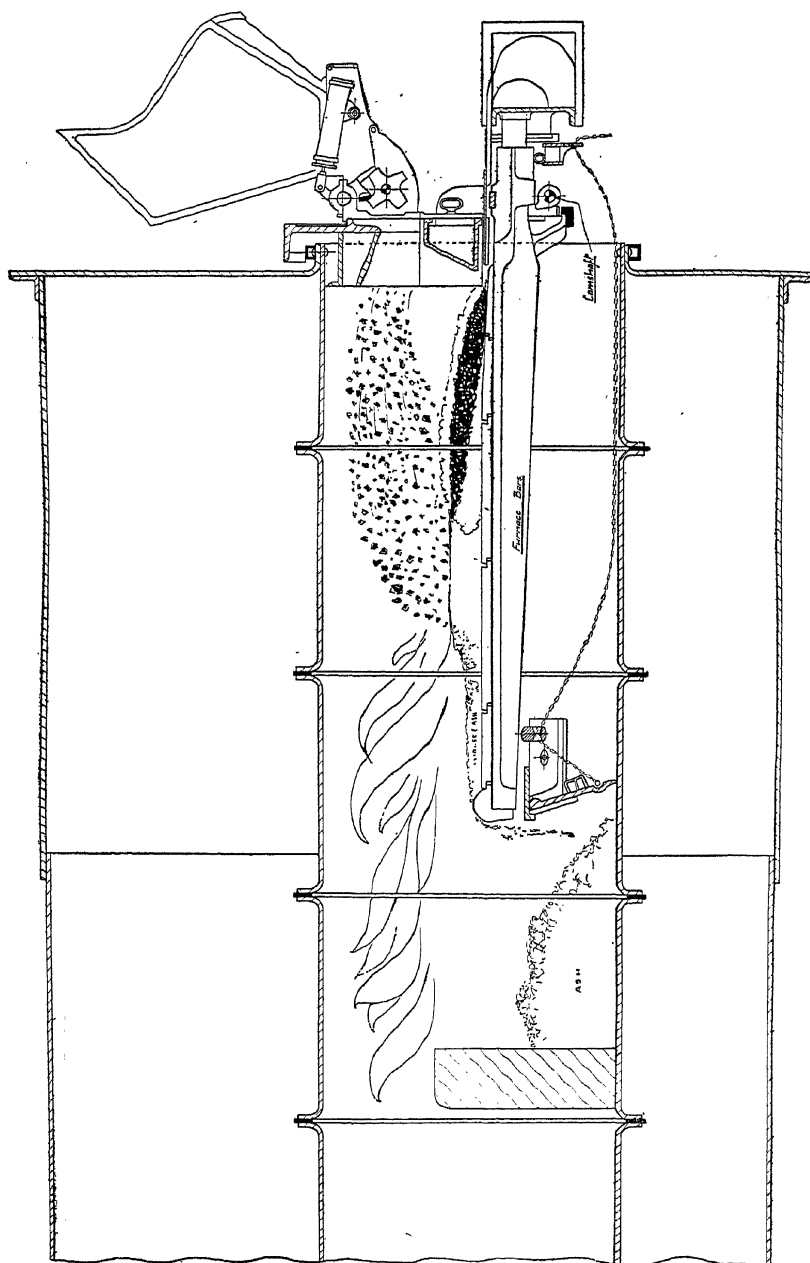


FIG. 82. Sprinkler stoker with variable cam mechanism.

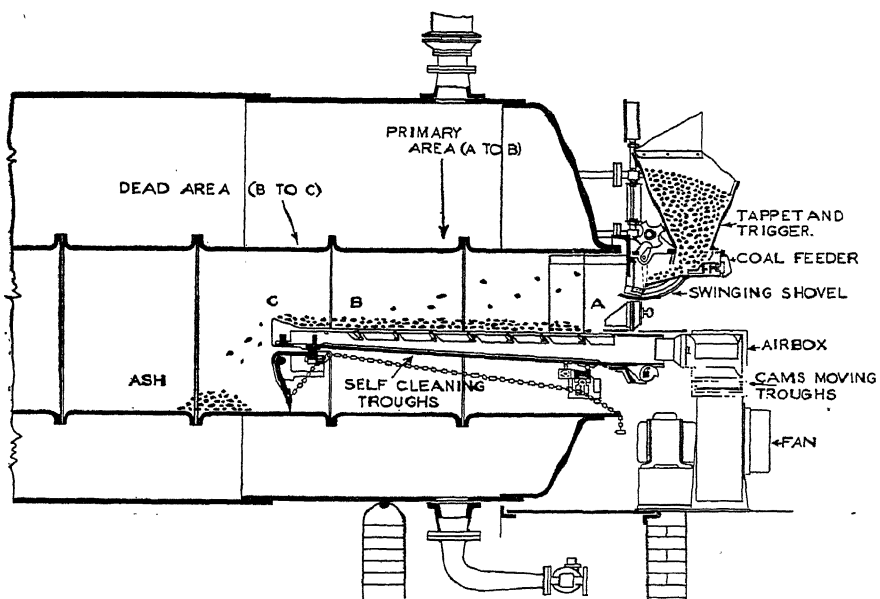


FIG. 83. Moving grate sprinkler stoker, shovel charging.

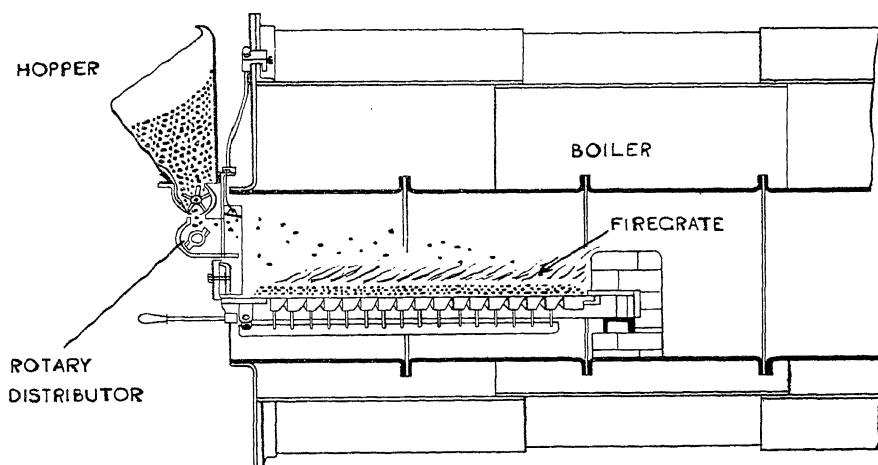


FIG. 84. Rotary sprinkler stoker.

firebars. If these defects occur they can be remedied by a few light strokes with the rake, care being taken not to disturb the whole firebed as this might lead to clinker formation.

Raking should be regulated by watching the  $\text{CO}_2$  content of the flue gases, which should be about 12 per cent. in the side flues. As soon as the  $\text{CO}_2$  content begins to drop attention to the fire or the draught is required.

Thinner and more even fires can be maintained with a rotary sprinkler, but care must be taken to avoid coal being thrown over the fire-bridge wall. The thinner fire answers more quickly than a thicker fire to variations in load, but it demands much greater care and must be maintained in very good condition, as otherwise it would burn out. For removing clinker sticking to the bars a flat slice bar should be occasionally used. Draught should be as low as possible. If the  $\text{CO}_2$  is low, probably too much draught is being used or the fuel bed is uneven.

When the boiler is off for cleaning the throw of the shovel or rotary distributor should be tested to see that it is delivering the coal on the right place.

With any fuel containing fines, grit emission is inevitable with all types of sprinkler stoker and a grit arrester may be required. At ratings higher than 30 lb. per square foot the emission of grits becomes serious and gets worse as the rating increases.

To achieve a rate of 30 lb. per square foot the draught required, measured halfway along the grate, would be as under :—

			Shovel type in. w.g.	Rotary type in. w.g.
Free-burning coals	..	..	0.4–0.5	0.25
Slightly caking coals	..	..	0.4–0.55	0.25
Medium caking coals	..	..	0.45–0.6	0.3
Strongly caking coals	..	..	0.65–0.85	0.3

There seems to be little difference in performance with this type of stoker on free-burning and caking coals, or between slacks and sized coals, though slacks require greater draught for the same burning rate. Both give similar  $\text{CO}_2$  content in the flue gases and similar output of steam, but there is a reduction in efficiency with fuels containing fines on account of grit emission as mentioned above, which increases at high rating. The excessive amount of steam frequently used to cool the bars or reduce clinker formation is unnecessary.

Under equal conditions, sprinkler stokers, as with all types of mechanical stokers, require less labour than hand firing, but do not obviate the need for the attention of the fireman even if the fuel is handled mechanically.

(2) *Coking Stokers.* Coking stokers imitate the coking method of hand firing, with the important difference that the fire-doors need never be opened except for banking. These stokers deposit the coal at the front of the grate from which position it is slowly moved to the rear by means of moving bars or rams (Fig. 85). The effect of this is to distil the volatile matter at the front portion of the grate from which it has a long travel through the hottest portion of the furnace and over the incandescent firebed, thus producing good conditions for securing complete combustion and obviating smoke.

The method of feeding the coal is such that the emission of grit is generally negligible even with coals containing fines.

The coking stoker does not answer so quickly to load variations as the sprinkler type.

The coal used should not be of a strongly caking character, but if such coal must be used the hard masses of coke that form must be broken up by means of the poker or by the action of the ram.

The standard thickness of fire is 12 to 14 inches at the front of the grate,

tapering toward the back (Fig. 85). Care should be taken to see that the fuel covers all parts of the grate. Bars too thinly covered, at the back will admit cold air, while if the fuel bed is too thick at the back, live coal only partly burnt may be carried over with the ashes. The cure when the end of the grate becomes uncovered is to increase the coal feed or reduce the draught; the cure for excessive carbon in ashes is the reverse; which method is adopted in each instance depends, of course, on the steam requirements. The back of this grate can be seen from the underside of the bars while the stoker is in operation, and if the bars are found to become red hot the probable cause is some factor which prevents sufficient air from passing through the bars to keep them cool, such as insufficient draught or the formation of clinker. The coal should be completely burnt before it falls off the end of the grate into the ash pit.

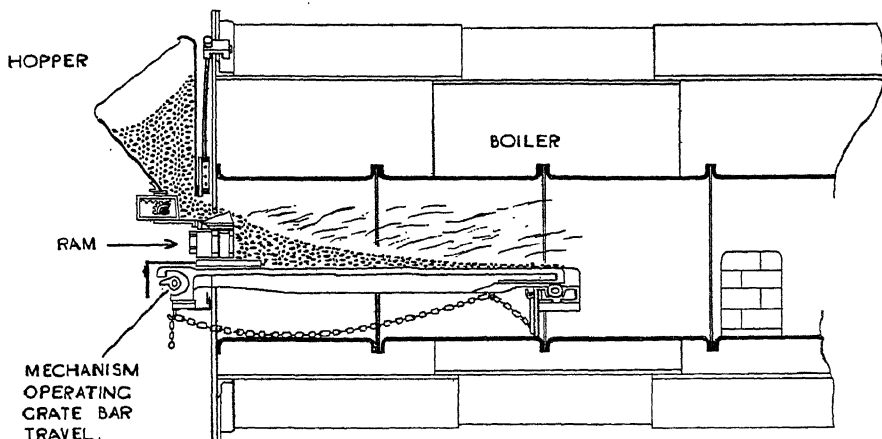


FIG. 85. Coking stoker.

This type of stoker requires more draught than the sprinkler, because of the greater thickness of firebed. The approximate draughts required in order to burn a variety of bituminous coals with induced draught at the rates stated are given below.

Burning rate lb./sq. ft. grate/hr.							Draught in. w.g.
20-25	..	..	..	..	..	..	0.25-0.3
25-30	..	..	..	..	..	..	0.4 -0.5
30-35	..	..	..	..	..	..	0.5 -0.6
35-40	..	..	..	..	..	..	0.7
40-45	..	..	..	..	..	..	0.8 -1.0

Apart from occasional slicing through the fire-doors, which are not opened for this purpose, this type of stoker requires very little manual attention; with some coals, apart from removing ashes and, with poor quality coals, breaking up the clinker, no labour is required. This, of course, does not mean that attention from a skilled operator is unnecessary. From the labour angle this type of stoker is claimed to require less manual attention, provided reasonably good coal is available for full load requirements, than the sprinkler type.

(3) *Underfeed Stokers.* Underfeed stokers have no counterpart in hand firing. They fall into two types:—



- (a) The pot type (Fig. 86), the larger sizes of which are also used for small water-tube boilers up to 25,000,000 B.Th.U. per hr. ; the smaller sizes are used for vertical boilers and C.I. sectional boilers (Chapter XXIV).
- (b) The grate type (Fig. 87).

The coal is fired mechanically from the bottom of the retort by means of rams or screws (cf. Fig. 29), the air for combustion being supplied by forced draught fans. As the coal rises in the retort, volatile matter is driven off and when conditions are right burns on the top of the fire with a short, intense flame.

In the pot type the ash falls over the brickwork down the sides and back ; in the flue type it falls towards the sides and remains on the grate. In both types it is withdrawn by hand.

The surface of the fire consists of a layer of incandescent burning coke. The

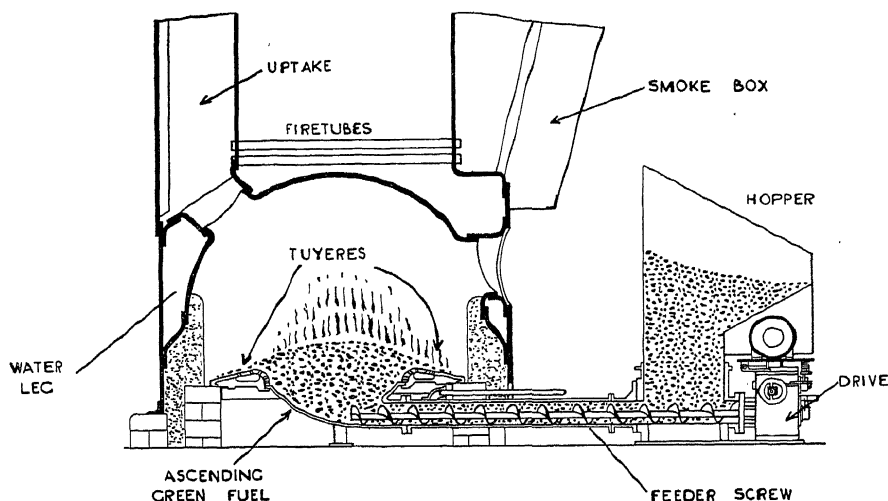


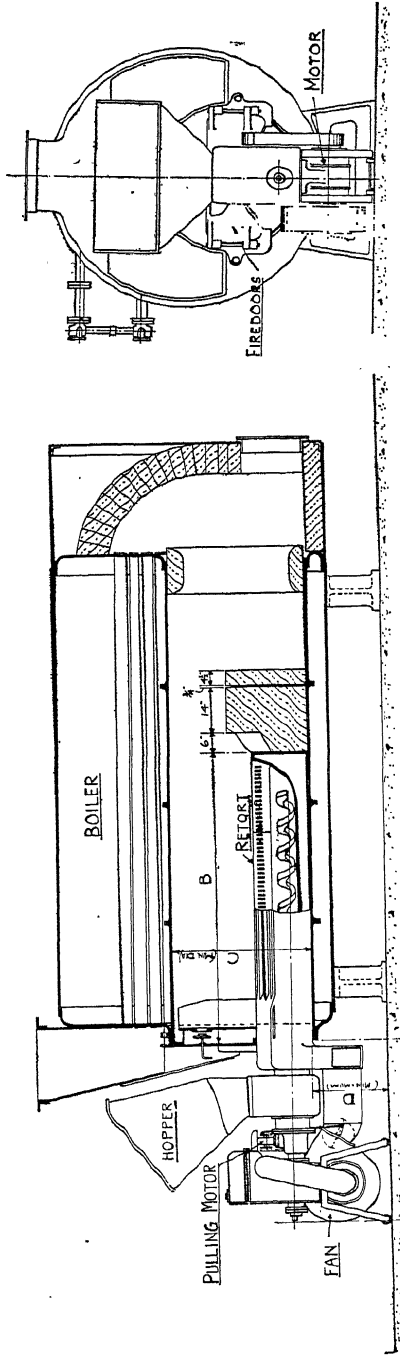
FIG. 86. Underfeed stoker—pot type.

air for combustion is introduced near the point where the fuel emerges from the retort or pot. As the volatile gases are liberated they are thoroughly mixed with air. The mixture passes up through the fuel bed where it encounters higher temperature zones and complete combustion of the volatiles occurs when the mixture passes through the intensely hot coke on the surface. All the air is introduced through the fuel bed from below. If smoke should be produced the air supply must be adjusted according to the analysis of the flue gases ; high  $\text{CO}_2$  means that more air is needed, and low  $\text{CO}_2$  that the air supply should be reduced.

The adjustment of the stoker must be such that coal emerges from the retort throughout its entire length (flue type), or over its entire area (pot type) ; when properly adjusted the fuel bed is automatically maintained clean as the ash is floated away from the retort.

These stokers are adapted for burning free-burning bituminous coals and semi-anthracites, but are more selective than the overfeed types.

II. *Stokers for Water-tube Boilers.* For water-tube boilers stokers fall into three main classes :—



Under the stoke

- (a) Progressive, in which the grate surface is movable and travels at the same speed as the coal which lies undisturbed thereon, e.g. the chain-grate type.
- (b) Non-progressive, in which the grate is usually fixed and the coal travels over the surface by gravity or otherwise, e.g. the retort type.
- (c) For smaller water-tube boilers some of the stokers designed for shell boilers are also used.

Modern stokers for water-tube boilers are highly efficient and very flexible, taking, as they do, a wide range of coals. Strongly caking coals with ash of low fusing point should be avoided for mechanical firing and obviously a stoker and setting which is designed for high-volatile coals will not operate satisfactorily on anthracite duff alone without considerable modification to the machine and the setting.

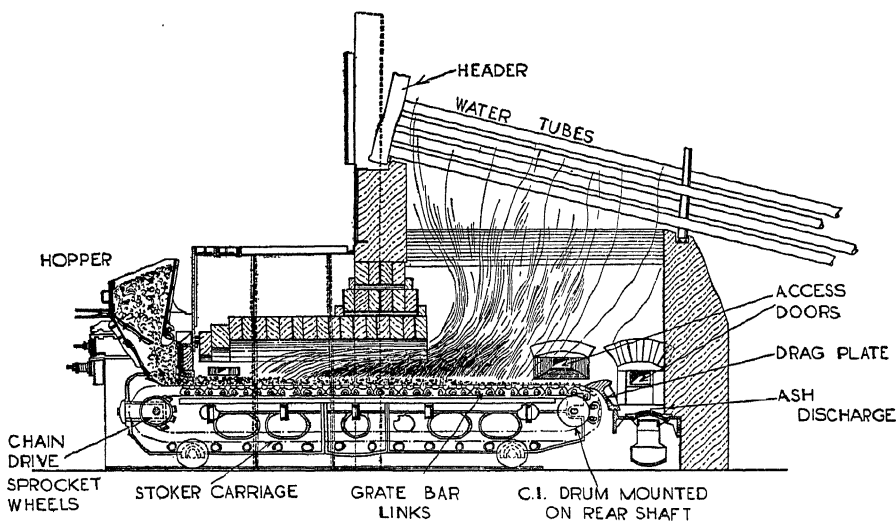


FIG. 88. Chain grate stoker.

A chain-grate stoker is shown in Fig. 88.

Full operating instructions are issued by the makers of stokers for water-tube boilers and it is impossible here to deal with the various adjustments for retort, natural, induced and balanced draught stokers, some of which are of the compartment type.

*Mechanical Stokers—General.* While any stoker will burn practically every coal with some degree of success, no stoker is a commercial success with every coal. Many of them are highly selective; if the boiler rating is to be obtained, so long as the use of unfamiliar and low-grade fuels must be regarded as a feature of boiler practice, it is strongly recommended that when adopting machine firing plant should be installed which is capable of dealing with the widest possible range of coals. It is not enough to know that on such and such a stoker low-grade fuel *can* be burned. It is necessary to know whether the required steam output can be maintained *with that plant and fuel*.

The importance of maintaining all types of mechanical stokers well lubricated and in good repair cannot be over-emphasised.

## WATER LEVEL

If the water level is maintained too high the steam passing from the boiler is liable to carry water with it. This water is usually discharged from separators, traps and drains and therefore represents a wastage. It is, therefore, important to keep the water as nearly as possible at working level by careful regulation of the feed pump. Excess of impurities in the boiler water aggravates priming.

Particular care must be taken never to allow the water level in the gauge glass to sink out of sight or get dangerously low. If this should happen and no water is blown out when the bottom gauge glass drain cock is opened, prompt action must be taken :—

- (a) Shut forced draught dampers and stop fans, but it may be advisable to leave the outlet boiler damper open.
- (b) Close all stop valves.
- (c) Smother the fires with earth, sand, ashes or even coal ; as a last resort, draw the fires.
- (d) When these steps have been taken, warn everyone in the vicinity of possible danger.

## BLOW-DOWN

Whilst a certain amount of blowing down is essential, if this is excessive or if the blow-down valves leak, avoidable heat losses occur. The amount of water rejected should therefore be carefully controlled and reduced to the minimum necessary to maintain the boiler water sufficiently free from impurities.

The pipes leading from the blow-down valve should be examined to see that no leakage is occurring when the valves are closed. A general indication that the boiler should be blown down is the surging of the water in the gauge glass.

Let  $x$  be the gallons of feed water added per day per gallon of boiler capacity,  
 $a$ , the total dissolved solids (T.D.S.) in grains per gallon in the feed water,  
 $y$ , the gallons of water blown down per day per gallon of boiler capacity,  
 and  $b$ , the T.D.S. in this blow-down.

Since to maintain the solids in solution at a predetermined level the amount of dissolved solids removed from the boiler at blow-down must equal the amount of dissolved solid added per day in the feed water,

$$ax = by, \text{ whence } y = \frac{ax}{b}.$$

Clearly, the total dissolved solids should be kept as high as is safe to avoid priming and the amount of blow-down will depend on this, on the rate of steaming and on the composition of the feed water (including any returned condensate).

## BANKING FIRES

Fires are banked for two purposes :—

- (1) So that the boiler can be restarted and brought up to full steaming quickly.
- (2) To give a continuous, though small, supply of steam over a period without attention.

The object must be to cause all the small amount of air that enters the boiler to pass through the hot part of the firebed so that it consumes the coal at a slow rate regulated by the main damper. Any air which does not pass through the hot part of the firebed comes in cold and leaves hot. The heat it acquires has been stolen from the boiler and is a direct waste of coal.

The bulk of the fuel should first be made red hot. The back of the grate is then cleared and all the good, live fire is pushed on to the back against the brick bridge. This fire is then built up with fresh fuel, any clinker on the

remainder of the grate being loosened, but left lying on the grate with the ash to help to prevent the passage of air through the front and middle of the grate.

All dampers are then closed to the minimum necessary to prevent the escape of fumes into the boiler house. Ash-pit doors and secondary air openings are also closed as far as possible.

The water in the boiler should be left 3 or 4 inches higher than the working level and care taken that the check valve, blow-off cock and water gauge drain cock are tightly closed, as otherwise the boiler may empty and damage result. Occasional inspection is desirable to ascertain that the water level is correct as shown in the gauge glass.

When restarting after the fires have been banked or after the dampers have been closed for any length of time, great care should be taken before opening the fire-door or breaking the fire to raise the damper and open the air regulators on the fire-doors. This is very important as it allows any unburnt gases which may have accumulated to be swept out of the boiler and thus avoids the risk of explosion and back-firing which at least are a danger to the fireman.

The clinker is then cleaned out and the live fuel drawn forward from the back. The fire is then gradually built up to the required thickness by firing lightly and often.

## CLEANING HEATING SURFACES

The smoke tubes of tubular boilers should be brushed out every day whenever possible to ensure the highest evaporation and the best efficiency.

To do this by hand methods the dampers should be almost closed and the doors of the boiler house shut if necessary so that cold external air may not impinge upon the hot end plate when the smoke-box doors are opened. The smoke-box doors may then be opened, one at a time, and the tubes swept with tube brushes while the boiler is under steam.

With good coal and good firing, tube cleaning can be done less frequently, but the tubes must be kept clean.

To avoid this hand labour, steam jet blowers are sometimes fitted whereby the cleaning is done by well-directed steam jets, and this system does not require the smoke-box doors to be opened and interferes in no way with the operation of the boiler.

Combustion chambers of return tube boilers should be cleaned out once a fortnight or as required.

In Lancashire and other brick-set boilers the flues and the outside of the shell within the flues should be swept clean as required and at regular intervals.

## FIREBARS

The space occupied by the solid portion of the firebars is not available for the passage of air, consequently firebars should be designed to give the maximum effective air space, and this should be as evenly distributed over the grate as possible. This does not preclude a gradual reduction from front to back, which is sometimes desirable.

Other essentials of firebars are :—

- (1) Resistance to overheating and, therefore, avoidance of constant renewals. This is promoted by the use of special types of cast iron of a heat-resisting character.
- (2) Such arrangement of the bars as will reduce to a minimum the amount of small coal falling through the grate.

Overheating can be reduced by so designing the bars that the heat which they receive from the fire is dissipated as rapidly as possible. This can be done

by reducing the area of absorption to a minimum and increasing the area of dissipation to a maximum, the area of absorption being that amount of metal directly exposed to the heat of the fuel bed, and the area of dissipation being that amount of metal exposed to the cooling effect of the incoming air. For example, a solid bar with an upper surface  $1\frac{1}{2}$  inches wide and only  $1\frac{1}{2}$  inches deep is more likely to become overheated and warped than a bar with an upper absorption surface of  $\frac{3}{8}$  inch and an elongated dissipating surface up to 5 inches, with or without additional heat dissipating surfaces in the form of fins.

The deep type of bar to which reference has just been made has the additional advantage that the space between the bars (available for air ingress) may be reduced and thus help in reducing to a minimum the amount of small coal falling through the grate. At the same time the number of spaces—although individually of lesser opening: say  $\frac{3}{8}$  inch for wide bars and  $\frac{1}{8}$  inch for narrow bars—is increased, resulting not only in a total overall increase of air space, but in much better distribution. This type of bar may be more expensive in initial cost.

A high burning rate per square foot of grate area tends in general to keep the bars cooler, because whilst a higher burning rate per square foot increases the temperature of the fuel bed, and therefore the heat received by the bars, this increase is relatively small compared with the greater cooling effect of the increased quantity of air sweeping the bars (cf. "Firebed cooling," Chapter VI). This would not apply to the same extent when preheated air is used. A high ash content in the coal will also tend to keep the bars cool. Conversely over-heating and maintenance charges increase rapidly with "ultra-clean" coal.

Attempts to overcome some of the disadvantages of the wide surface type of stationary firebars have been made by making them hollow.

Where stokers are designed to work with moving bars—which mechanically reject the ashes into a back ash pit—the lower limit of width of bar is limited by this width being able to provide a cam strong enough to work with the propelling mechanism. Where this type of moving bar is employed its functions are said to be enhanced either by placing notches in the bar to give a more positive action to the ash removal, or by giving them a reciprocal action which is particularly useful in breaking up a strongly caking fuel. The correct relative movement of the bars may be an important factor in satisfactory combustion.

Forced draught can be applied either by closing the chamber under the bars or by the use of troughs into which either fixed grids or small loosely fitting bars are inserted.

## CARBON LOSSES

In the operation of any stoker apart from reaching the best combustion conditions as previously described, losses must be reduced to a minimum. These losses are:—

- (1) Loss of carbon in grit emission.
- (2) Loss of carbon in riddlings through grate.
- (3) Loss of carbon contained in the ashes.

(1) *Loss of Carbon in Grit Emission.* This source of loss is of relatively recent occurrence and arises out of the modern practice of burning coal at high rates of combustion per square foot of grate area. Apart from the nuisance which it causes to the surrounding property and actual danger in the damage to eyes of personnel and parts of machinery, it involves what may be an appreciable loss of carbon.

The amount of grit emitted depends, in the first instance, on the rate of combustion, i.e. the intensity of the draught, and it may be further influenced by various other factors, as follows:—

- (a) The size of the coal.

The emission will be greater from slack than from sized coal.

- (b) The type of coal.

The emission will generally be greater with a free-burning coal than with a caking coal.

- (c) The type of stoker.

The emission will tend to be greater with the sprinkler type of firing than with the coking type, because with the sprinkler type the very fine particles are more readily caught up in the air stream as the coal is flipped on to the grate. Some alleviation may be obtained by thoroughly wetting the coal, which causes the smallest particles to cohere; this also applies to hand firing. The quantity of water added should be just sufficient to make the fuel "ball" when pressed in the hands.

- (d) The operation of the stoker.

As far as possible disturbance of the fire should be avoided as this must necessarily tend to increase grit emission.

The emission of grit will tend to dirty the boiler and reduce the periods between cleaning. It will also erode induced draught fans.

(2) *Riddlings*. In all grates there is a tendency for small coal to fall through the bars. This tendency increases with free-burning coals and coals with a high percentage of fines. It is particularly prevalent in stokers with moving parts. It may be due to the bars being too widely spaced for the fuel burned or the actual movement of the bars. The riddlings should be returned to the coal in as uniform quantities as is convenient.

Excessive riddling may occur with chain-grate stokers if the lateral working clearance between the links is allowed to become too great. The clearance should be maintained at about the design figure, which with one type of chain-grate stoker is approximately  $\frac{1}{8}$  inch per foot width of chain.

Again, with some types of chain-grate stoker, the bottom section of the hopper front consists of a hinged drag plate which rests on the chain. When in good condition this makes contact with the chain where the links have closed again after passing round the front sprocket. In the process of time this plate is liable to wear and assume a steeper angle, the bottom edge of the plate being advanced to the point where the break of the chain is not covered, so that small coal falls through direct into the riddling pit. These plates should always be maintained in good condition.

(3) *Loss of Carbon in Ashes*. The separation of clinker and ashes from the fuel bed invariably involves the entrainment of a certain amount of coke which, under bad conditions, may amount to 5–10 per cent. of the coal burned. The amount will tend to be greater when coals of high ash content are used.

The amount frequently depends on the skill of the firemen. A certain loss is unavoidable, and steps must be taken to recover as much as possible. To obtain a rough idea of its magnitude, an average sample of ash and clinker should be quenched in water immediately after leaving the fire, and the pieces of coke picked out by hand; the proportion may be more exactly ascertained by chemical analysis.

With a properly operated furnace using suitable coal, the ashes would not contain excessive amounts of unconsumed carbon. The first step, therefore, should be to investigate combustion conditions at the furnaces in order to reduce carbon losses to reasonable proportions. The chief causes of unconsumed carbon in ash are given below with some notes upon each of them.

- (1) The coal is unsuited to the grate and the furnace.

Too frequently this is regarded as the prime cause of the trouble, and other causes are not investigated. There are circumstances—as for example when

the coal is too small or too large or contains quite excessive amounts of ash—under which the coal is the cause of the trouble. Generally, however, means can be found for burning most coals with reasonably satisfactory results.

This applies both to hand firing and mechanical firing, but more particularly to mechanical firing.

- (2) High content of ash fusing at a low temperature.
- (3) Too high a fuel bed temperature fusing ash to clinker.
- (4) Too frequent cleaning of fires due to clinker.

Causes (2), (3) and (4) all arise from clinker formation. If clinker formation is excessive carbon particles will be enclosed in the clinker and thus cannot burn. The remedy is to prevent the clinker from fusing. Clinker formation takes place at the zones of highest temperature. When the fuel passes down the fire undisturbed at the zone of highest temperature so much unburnt material will be present that the ash particles will not be in contact with one another to any considerable extent; large masses of clinker cannot normally form under these conditions unless the ash is very fusible. If the fuel bed is disturbed in such a way as to bring the bed of ash lying on the grate upwards into the combustion zone, conditions are immediately created which favour clinker formation. The fuel bed should not be poked or disturbed in such a way as to create these conditions.

Clinker is more readily formed from smalls than from larger sizes of graded fuels, because smalls are massed together and the ash released is in closer physical contact. The size of the ash particles from small coal is also smaller, which again leads to easier fusion. The character of the ash is therefore as important as the quantity formed.

If the ash fusion temperature is so low that even with the precautions just mentioned clinker is formed, a remedy that has been found satisfactory is to keep the ash bed cool by admitting steam or very finely atomised jets of water under the grate. The cooling medium should be focussed upon the hottest zones where trouble is likely to originate (cf. Chapter VI).

- (5) Excessive use of the poker, slice bar and hoe, when hand firing.

The comments made under (2) above will illustrate the importance of not disturbing the firebed too frequently and thus bringing the smaller pieces of coal on to the grate before they have been burnt.

- (6) Careless stoking resulting in the fuel bed not being kept level.

This refers primarily to hand firing, but mechanical stokers should be watched to see that defects do not occur in the fuel bed.

- (7) Firing coal on to bare spots on the grate.

Items (6) and (7) are special cases of the same cause, bare spots on the grate being the extreme condition arising from an uneven firebed. Obviously, if fresh coal is fed directly on to an uncovered grate the grate will serve as a screen through which the smaller pieces of coal will drop.

- (8) Furnace volume too small for rate of combustion desired.

This difficulty should not arise with furnaces or boilers operated at the load for which they were designed. The difficulty arises principally when the furnace is pushed to give an excessively high output.

When attention to operating details fails to reduce the quantity of carbon in ashes to a negligible amount, consideration should be given to recovery for re-burning.

The following are methods of recovery :—

*Hand Picking.* The amount and size of the coke pieces frequently warrants hand picking.

*Forking.* Where supervision is good, large pieces of coke are rare, but the smaller material may contain a lot of carbon. The residue should be picked



over with forks having  $\frac{3}{4}$ -inch spaces between the prongs, the undersize being returned to the fire and the oversize being discarded. In larger installations this may be done mechanically. Some ashes contain their carbon in the form of large material and others of small material. The distance between the prongs of the fork should be selected in accordance with the characteristics of the material to be treated.

*More Elaborate Recovery Methods.* Some of the more elaborate methods are based upon the principles of coal washing already described in Chapter I. The ashes and clinker are broken up and the carbonaceous portion is floated off either in a clay-water suspension or in calcium chloride solution or in mechanically agitated water. A good deal can frequently be done by floating in water if the clinker is dry, since the highly porous particles of coke are generally sufficiently light to float in water for a short time until the water has entered the pores. Magnetic methods are also used on the principle that a good deal of the iron and iron oxides present in the clinker is magnetic, whereas the coke is not.

### FORMATION OF CLINKER IN FUEL BEDS

The formation of clinker in fuel beds and the methods for the alleviation of this difficulty have been discussed at some length in Chapter VI. It should be noted that it is the character of the ash and not its total quantity that determines the difficulties arising from the nature of the clinker. When a low ash coal gives a bad clinker the trouble can sometimes be remedied by increasing the ash content either by adding shale or sand. It is, for example, quite customary to add broken firebrick to locomotive fires.

## CHAPTER XII

## PULVERISED FUEL

*Equipment and application :—Pulverisation—Extraction fans—Distribution systems—Burners—Application to boilers—Cement—Metallurgical applications in the ferrous and non-ferrous industries—Choice of fuel—Furnace design.*

*Efficient use of P.F. in boilers :—Differences between operation on pulverised fuel and firing on grates—Bogey results—How to survey a pulverised fuel boiler installation.*

## INTRODUCTION

THE combustion of coal in the form of nuts, etc., differs from the combustion of gases in the extent of the surface of contact. A mixture of a combustible gas with air achieves molecular contact so that the surface exposed is the maximum and combustion can be so rapid as to be

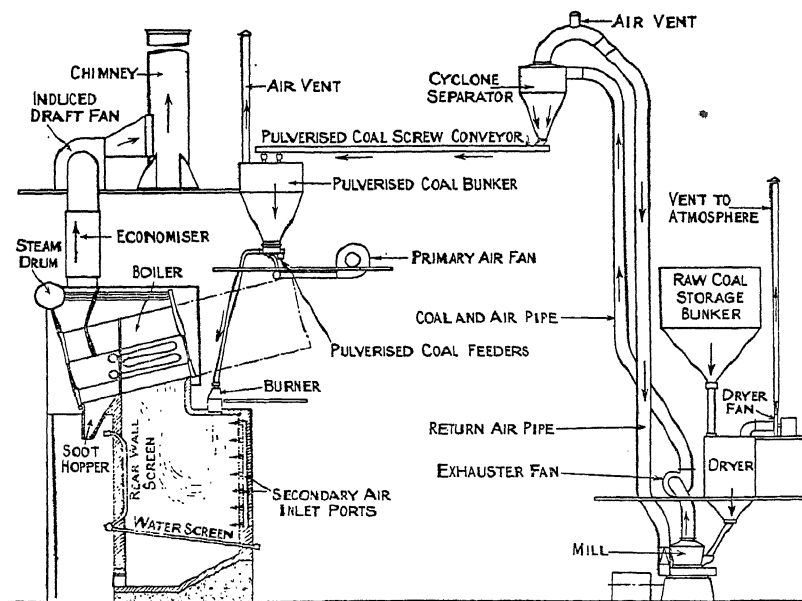


FIG. 89. Typical bin and feeder system.

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explosive. The surface of large coal and even of slack is limited by comparison so that combustion is slower. Liquid fuels are fired in a finely atomised form in order to secure the greatest possible surface of contact between fuel and air.

A similar method can be applied to coal (Fig. 89). By reducing the coal to a fine powder, which is then carried forward by an air blast, it is possible to obtain perfect combustion with absence of smoke when using only a small excess of air. High temperatures and high efficiencies result from this practice when correctly applied.

The earliest attempt to utilise pulverised coal was made in 1818. Various attempts were made between then and 1895, when the first really successful

application was made, this in the firing of cement kilns. Little progress was made following this till about 1918. From then on progress was rapid. The increase in the use in the 10 years following amounted to over twentyfold. By this time it had been demonstrated that all classes of solid fuel could be successfully pulverised and burned in that form.

Combustion of coal takes place in two steps, the first being the separation of the coal into gaseous hydrocarbons and coke, the second being the combustion of these fuels.

In the ordinary coal-fired furnace the volatile matter is driven off and burned as the coal is fed on to the grate, and the coke left behind then combines with the necessary air. In using pulverised coal the fuel, mixed with the correct amount of primary air for ignition, is injected into the furnace in a finely divided state. Under this condition the distillation of the volatile matter takes place very rapidly, and owing to the small size of the solid particles of coke for all practical purposes the result is one stage combustion.

The coal must be freed from most of its surface moisture (though it may still contain all its inherent moisture) and is then pulverised to the required degree of fineness. Practically any dry fuel, whether high or low in ash content or volatile matter, can be burnt successfully in pulverised form, but the most suitable coal contains over 20 per cent. of volatile matter and is of not higher than medium caking power. With lower volatile content it becomes rather more difficult to burn completely: low-volatile coals require large combustion spaces if complete combustion is to be obtained. The time taken for complete combustion, which in turn governs the size of the combustion chamber, depends on the fineness of grinding. The type of coal used will depend also on the nature of the process to which pulverised fuel firing is applied (see later).

The following pages contain first a description of plant and technique and later a discussion of the practical problem of burning pulverised fuel to the best advantage.

## EQUIPMENT AND APPLICATION

### MACHINERY USED FOR PULVERISING

The ball mill requires slightly more space than the higher speed mills. It is generally somewhat more sensitive to moisture in the coal, and it is not so flexible as those of higher speeds because the no-load power taken by this type of mill represents a fairly high percentage of the full-load power. It therefore follows that the power on low loads is proportionately higher. The speed of operation is generally between 20 and 30 revolutions per minute.

The slow speed type of machine consists in one typical design of a drum revolving on hollow trunnions, the drum being partially filled with steel balls of various sizes. The drum is revolved at the correct speed to lift the balls almost to the top centre, from which they drop in a cascade on to the (pre-crushed) coal to be pulverised. The coal is fed through one of the hollow trunnions and delivered from the hollow trunnion at the opposite end.

The power consumption of this type of mill, together with its exhausting fan, when grinding to the following specification:—

	Per cent.
Residue on 200 B.S. sieve .. .. .	15
Residue on 100 B.S. sieve .. .. .	3-5
Residue on 30 B.S. sieve .. .. .	·05-·1

and working at its full rated load, is in the neighbourhood of 24 kWh per ton of coal pulverised, and varies according to the size of the machine, the type of coal being pulverised and the moisture content of the coal.

The medium-speed machines, i.e. those operating between 800 and 1,600 revolutions per minute, of the hammer type, and also of the combined hammer and moving peg type, owing to their design and speed, are more capable of dealing with coals of a higher moisture content. A drying system is often incorporated as an integral part of the machine. The necessary heat is either drawn

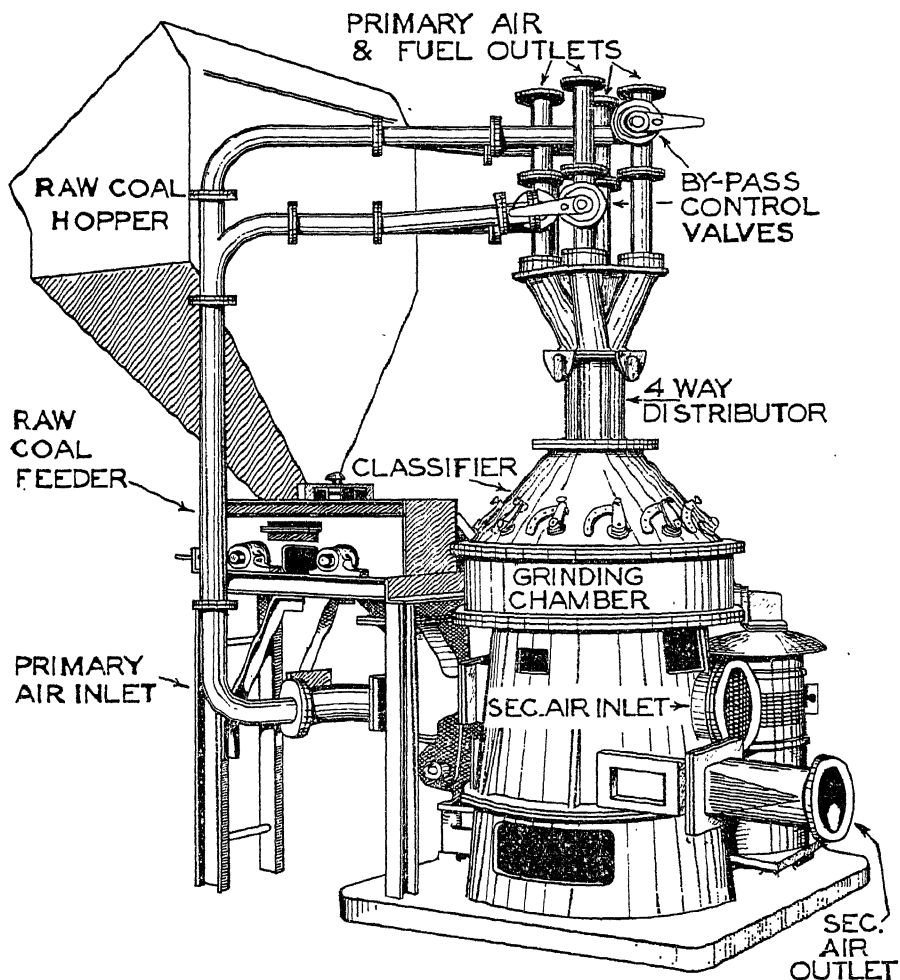


FIG. 90. Typical pulverising unit.

from an air heater or from hot waste gases. A self-contained unit of this type is capable in many instances of dealing with moistures up to 12–14 per cent.

There is a much greater difference between their full-load and no-load power requirements than in the slow-speed machines, and therefore they have a somewhat more flexible power curve for grinding over a wider range. The power consumption taken on full load is not very dissimilar to the ball mill, however, and under equal conditions would again be in the neighbourhood of 24 kWh per ton of coal pulverised.

The foregoing remarks are equally true of the high-speed mill, but it should be borne in mind that the very high speed mill tends towards a higher maintenance cost. All types of machines have an economic speed as regards maintenance. These high-speed mills sometimes run up to speeds of nearly 3,000 revolutions per minute.

Certain medium-speed machines of the combined hammer and moving peg type are capable of dealing direct with coals up to 2-2½ inches in size. The moisture content of this size of fuel is often lower, due to the fact that the surface exposed per lb. of coal, as compared with say ¼ inch peas or beans, is less. Slow-speed mills are less readily capable of dealing with coal over ½-inch in size.

A typical pulverising unit is shown in Fig. 90.

It is thoroughly sound practice to install pulverising equipment with white-washed walls and floors, and the house should be well illuminated. In this way it is readily possible to detect dirt. A well-kept plant is likely to involve low maintenance costs, but if the pulverising plant is installed in a dark and dingy hole with little illumination, the plant is not so likely to be well looked after, and small faults such as leaks are not detected before they multiply, and bring a train of other troubles in their path.

### FINENESS OF GRINDING

Opinions vary as to the degree of fineness to which the coal should be ground. The size of sieve must be clearly specified. A comparison of the more commonly used I.M.M. and B.S. meshes is as follows:—

Sieve	Size of aperture
100 I.M.M.	·005 inch
100 B.S.S.	·006 „
200 I.M.M.	·0025 „
200 B.S.S.	·003 „

The function of the pulveriser is to grind the fuel as finely as possible with a minimum of power. The burner, on the other hand, should be designed to burn efficiently the coarsest possible coal, but complete combustion is more readily obtained with a fine state of division of the coal. Ultra-fine grinding must result, however, in higher power costs, and a marked increase in wear. Again, the highest standard of grinding may not always be necessary. It may tend to give a flash heat rather similar to oil firing. When firing heavy ingot furnaces, for example, such conditions are not considered desirable.

The question of fineness should be governed by the requirements of the particular application concerned. For instance, when firing large water-tube boilers, 65 per cent. through 200 B.S. mesh is regarded as adequate, care, of course, being taken to eliminate the coarse particles on the 30 mesh. On the other hand, when firing Lancashire boilers there is a clear case for fine grinding. Here firing is virtually effected down a water-cooled metal tube which is a very confined space compared with the large combustion chamber of the modern water-tube boiler.

Similarly, on very small drop stamp furnaces, of which many are being fired with pulverised coal, there is again a call for finer grinding than that required for large ingot heating or continuous reheating furnaces. Generally speaking, more emphasis should be laid on *uniformity of grinding* than fineness of grinding. If one could be sure of obtaining 100 per cent. through the 100 B.S. mesh, there would be fewer difficulties, since this would automatically ensure freedom from large particles, and there would still be a sufficient number of fines on the 200 mesh.

It will be evident from the above discussion that hard and fast rules of

practice have not yet been laid down, but for general furnace work the specification previously given under " Machinery used for pulverising " will be found suitable.

## EXTRACTION FANS

In most types of pulveriser, extraction or suction fans are used for removing the pulverised coal and delivering it either direct to the furnaces or to the storage bins.

The pulverised coal fan, termed the primary air fan, when used on direct firing systems is generally of the simple paddle-wheel type, the scroll of the fan being made of sections of either chilled iron or reasonably heavy mild steel, these sections being easily removed for replacement when wear has taken place from the abrasive action of the coal. For the same reason the vanes of the rotor are generally made of either mild steel or chilled iron, and also easily removed for renewal purposes.

In the bin or storage system the coal is usually fed by means of screw feeders placed underneath the storage bin. This device consists of a simple screw rotated inside a metal casing. The coal thus fed is not delivered to the suction side of the fan, but to the delivery pipe leading from the fan to the burner. If necessary the air pressure in the delivery pipe can be reduced to below atmospheric at the point of entry of the coal by providing a correctly designed Venturi tube in the fan ducting and admitting the coal at the throat of the Venturi tube. Thus this fan may be of ordinary construction without any necessity for special precautions against abrasion required for primary fans which have to handle the coal.

## DISTRIBUTION SYSTEMS

There are three main systems applying to pulverised coal, as follows:—

(1) *Unit System.* One pulveriser is applied direct to one or two furnaces. The application does away with all bins, and is suitable for medium sized and isolated furnaces. The size of pulveriser with this method must be selected carefully, otherwise the air for sweeping the mill, i.e. for the mechanical process of grinding, will not always coincide with the quantity of air necessary for combustion. This is often the simplest application, but it has its limitations. It would be impossible for example to cart coal to each of a large number of furnaces cramped together in a confined space.

Typical examples of an installation of this type are a single pulveriser firing direct to either one Lancashire boiler or one water-tube boiler; and one small pulveriser firing a continuous reheating furnace or a forge furnace.

(2) *Bin and Feeder System.* This is usually the most complicated type of installation. It consists of a suitable pulveriser, of the slow-, medium- or high-speed type. A slow-speed pulveriser will generally have a separate coal drying plant (Fig. 89).

From the pulveriser the coal is delivered by means of a fan to a bin above which is a cyclone. This cyclone separates the air carrying the coal from pulveriser to cyclone, and deposits the coal in the bin. At the base of this bin are a number of feeders generally of the screw or rotary drum type, each distributing the pulverised coal to a boiler or furnace.

From the feeder the coal drops into an air stream supplied by a primary air fan, which delivers it through ducts to the burner.

This system is very suitable for distributing to a number of points in various directions, and has the advantage of centralising all the coal handling at one point. The number of feeders that can be put on one bin is limited. The more feeders on the one bin, the larger it must be, and the more sensitive the installation becomes to the influence of certain disturbing factors. Amongst these is

the tendency for moist coal to hang up in the bin, for the reason that its mass is increased whilst the rate of flow of coal through the bin is decreased. While large storage bins have their advantages, there is much to be said for keeping the bin size as small as is reasonably possible.

A typical example of such an installation is a central plant firing a battery of boilers or a battery of furnaces; the furnaces may range in size from large continuous billet or ingot heating furnaces, or forge furnaces, to small stamping furnaces.

(3) *Ring Main System.* The coal preparation plant, inclusive of the pulveriser, is similar to that used in the bin and feeder system, in that the coal is pulverised and delivered by fan to the cyclone, which in its turn precipitates the coal into the pulverised fuel bin below it.

At the base of this bin are a number of feeders, each feeder supplying coal to a ring main which travels round the shop in which are the furnaces or boilers to be fired, and returns any surplus coal back to the cyclone for re-distribution. The mains may vary in length from 200 feet to 1,000 feet.

This system is particularly suitable for firing batteries of medium-sized furnaces with fuel consumptions each of, say, 30 to 350 lb. per hour. Its capital cost is considerably less than that of the bin and feeder system, and some 15 furnaces can be fed from one feeder point. By having four feeders at the base of a bin, each serving one ring main, it is possible to deal with a total

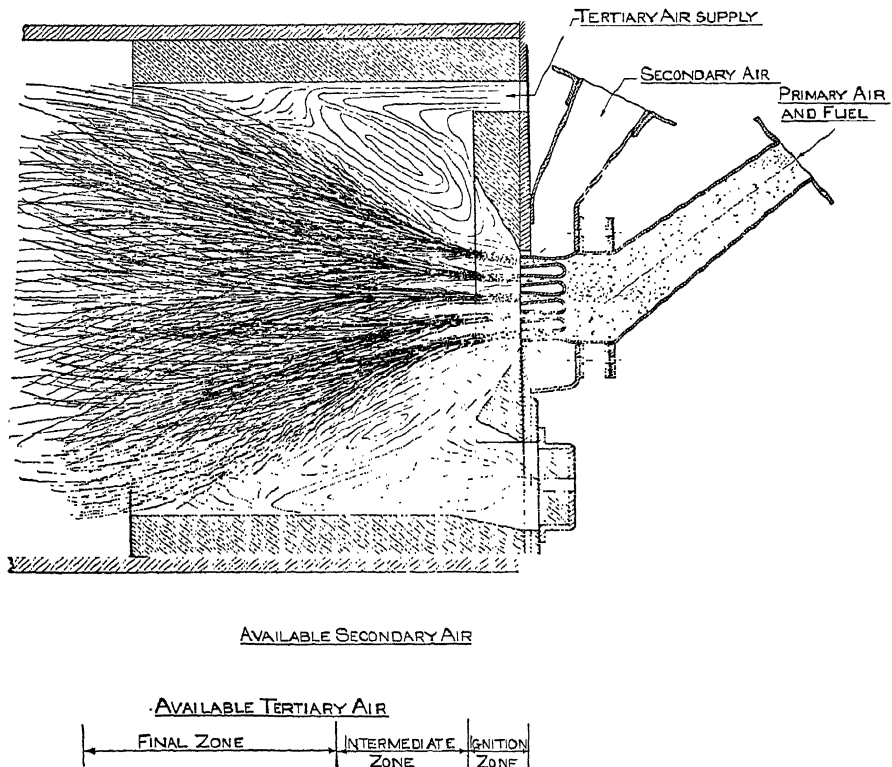


FIG. 91. The Fuel Research Board's grid burner.

of 60 furnaces. Obviously, to put down a bin with 60 feeders would be out of the question, firstly on capital cost, and secondly because of the size of bin which would be entailed. The only other method to adopt under these conditions is to have a bin and feeder system which in its turn is required to feed coal to secondary bins, each of which carries its own battery of feeders. The method is, however, less satisfactory than that involving the use of the ring main.

## BURNERS

Burners for large water-tube boilers with combustion chamber capacity perhaps of the order of a 20-foot cube, should obviously have wide dispersive qualities. They are often of the semi-rotary type where the coal stream is given a rotation in one direction and the secondary air in the other, the idea being to obtain an intimate mixture which is widely dispersed throughout the chamber.

However, exactly the reverse may be required, for example, in a long narrow bar furnace for heating drop stampings. An intimate mixture of the air and coal is again required, but wide angles of dispersion which would merely fling the coal particles on to the brickwork are detrimental.

These two examples are given to illustrate how varied may be the requirements of different burners for different purposes.

The velocity of the primary air stream and coal leaving the burner generally varies between 30 and 60 feet per second according to the particular application concerned. The secondary air velocities are of the same order, but vary according to the loading of the burner. Generally, the primary air stream is constant and as the quantity of coal is increased or decreased the secondary air is altered as necessary. The velocity of the secondary air thus varies in order to obtain correct combustion as more or less coal is introduced into the burner.

The Fuel Research Board's Grid Burner is shown in Fig. 91.

## GENERAL

At one time it was widely believed that pulverised coal firing was simply a means of burning dirty coal which was unusable by other methods. Fortunately this idea has given way to a realisation that pulverised coal is a highly scientific method of utilising to the highest extent the heating value of a given coal.

With this object in view, benefit results from the use of the cleanest and driest coal which is economically practicable. Where coal having a high ash content must be used much can be done to obtain the best possible results by pulverisation to a maximum degree of fineness.

The lower the moisture content the better. With preheated air most installations can handle coals containing as much as 10 per cent. of moisture, but such a moisture content will reduce the steaming load obtainable, and is best avoided where possible.

Practically all types of boiler, from the power station water-tube type down to the small vertical heating boiler, have been fired successfully with pulverised coal.

In water-tube boilers it is found practicable to obtain an overall efficiency up to 80 to 85 per cent. ; and the Lancashire boiler can be made to give from 70 to 80 per cent. when fired with a good quality of pulverised coal, this figure including the added efficiency of superheaters, economisers and air heaters.

There are many types of furnace operation in this country to which pulverised fuel firing has now been applied. The flexible nature of the method of firing, and, where batch heating is involved, the fact that the source of heat is stopped immediately the operation is completed are conducive to economy.



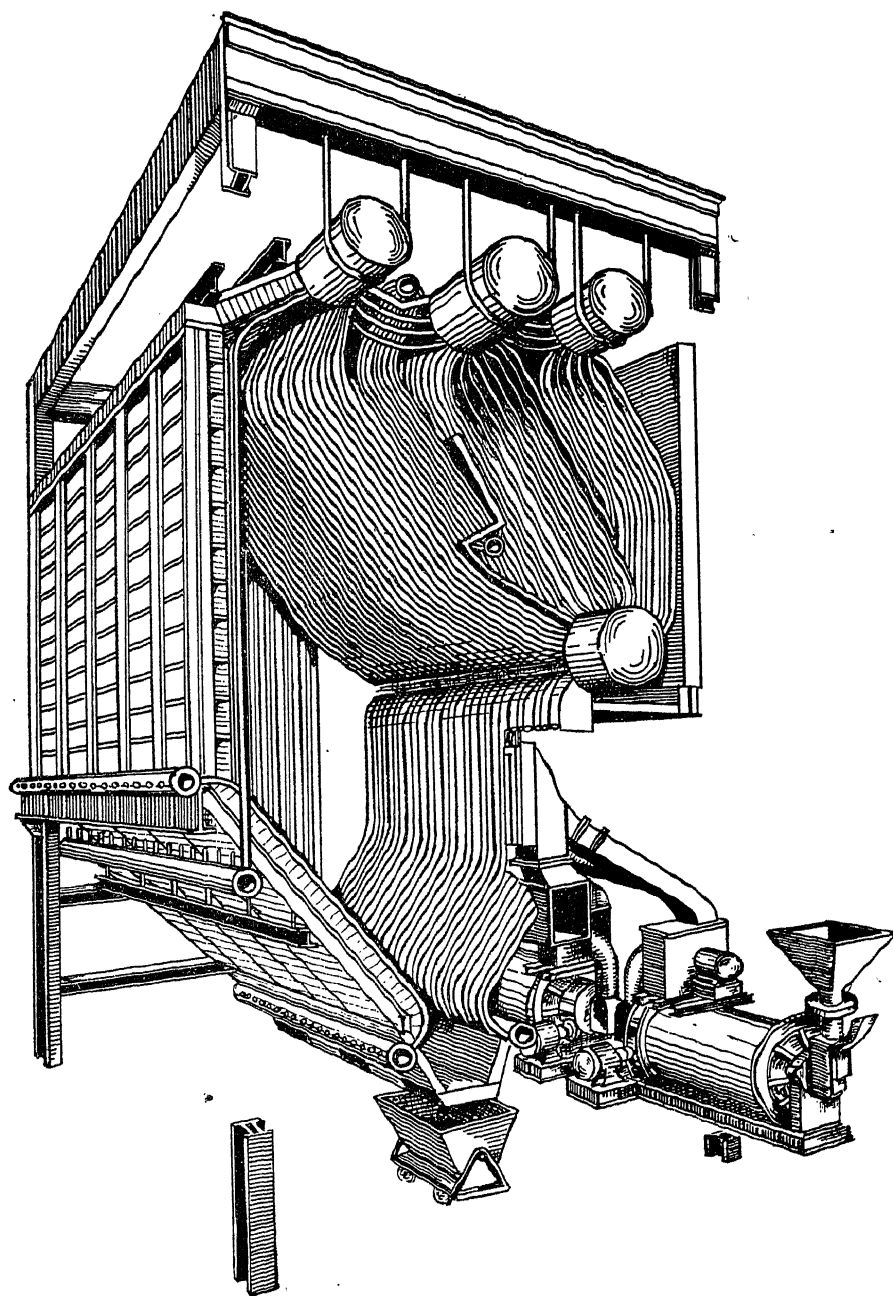


FIG. 92. Powdered fuel firing applied to a water-tube boiler.

For the control of fuel-air ratio the fineness of the coal should be checked periodically by taking small samples of say 100 grams and passing these through a 200 B.S. mesh sieve. By shaking and tapping this screen all the coal of the correct size will pass through the screen, and the balance which will be held on the screen should then be carefully weighed, thus determining the percentage of the coal which is reduced to the correct size. The volume or weight of air which is being delivered by the fans is checked by means of comparatively simple instruments (Chapter X). The volume of air delivered by the various fans may be controlled either by dampers placed in the suction or delivery ducts or by variable speed motors.

As with other methods of firing solid fuel a  $\text{CO}_2$  recorder will serve to verify the combustion. A steam flow meter is also of great help as the plant operator is able, with a little experience, to learn the control settings at which his fans should operate in conjunction with varying rates of steam flow.

The applications for pulverised coal firing fall into three main categories—boilers, cement and metallurgical.

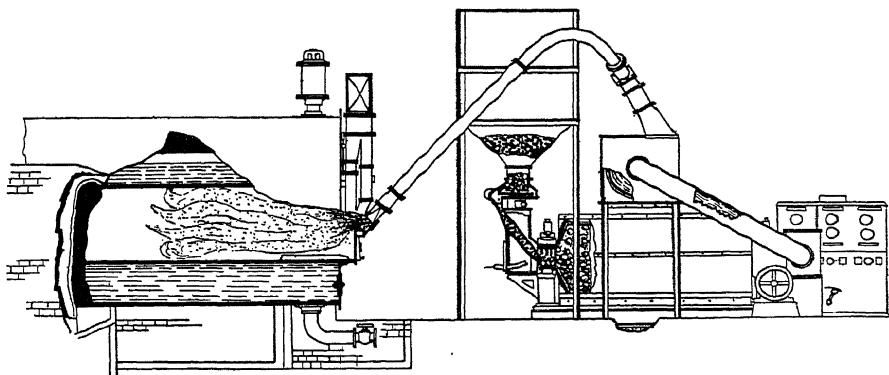


FIG. 93. Powdered fuel firing applied to a Lancashire boiler.

### BOILERS

The application of pulverised coal to boiler firing has long been established practice. The main applications are to water-tube boilers (Fig. 92), but of late years a number of Lancashire boilers have been fired with success (Fig. 93). The application to economic and small vertical boilers down to a rating of 1,000 lb. per hour steam evaporation is by no means unknown.

The application of powdered coal solely to a single small vertical boiler of 1,000 lb. per hour steaming capacity, might not be generally economical, unless the boiler were only one point to be fired in a large installation, then the position would be entirely different. Nevertheless local circumstances may alter the position as powdered coal is used successfully on small heating boilers.

### COMBUSTION CHAMBERS

Combustion chambers may be of several types depending on the duty required.

- (1) They may be constructed of solid brickwork lined with refractory bricks.
- (2) They may be of similar construction, arranged with an air space between the inner refractory bricks and the outer common brick wall.
- (3) They may be of the suspended wall type, with the hollow space again between the refractory and the outer wall.

- (4) The furnaces may be completely or partially equipped with what are known as water walls, these consisting of boiler tubes connected to the main boiler on their upper ends, and to headers outside the combustion chamber on their lower ends.

It is always advantageous to use preheated air both for primary and secondary air supply. This air may be heated by air-heaters placed in the flues beyond the combustion chamber or by means of the hollow walls just mentioned.

Users of pulverised fuel should investigate their plants with a view to discovering any sources of waste heat which might be utilised as a means of heating air. Economy and ease of operation could be improved in this manner.

In new installations where new or reconstructed combustion chambers are called for, it is recommended that wherever practicable the walls be of hollow construction. Air can then be drawn through these walls by the primary air fan or by means of a small subsidiary fan, and whilst this will give valuable preheated air for combustion it will also serve to prolong the life of the brickwork and render working conditions more pleasant for the personnel.

### **CEMENT INDUSTRY**

This industry was one of the earliest and is now one of the largest users of pulverised coal. Generally speaking the powdered coal stream is introduced through one burner at the end of a slowly rotating kiln, the dimensions being approximately 10 feet internal diameter and 240 feet long. Further information will be found in Chapter XIX.

### **METALLURGICAL APPLICATIONS**

#### **ANNEALING FURNACES**

Powdered coal is being used successfully for the annealing of steel and malleable iron castings. This latter application is relatively simple and may be relied upon to give a satisfactory anneal, with low fuel costs. The ring main method of distribution is particularly suitable for firing a large number of such ovens in any given works.

#### **SMALL FORGE FURNACES AND DROP STAMP FURNACES**

Many hundreds of these furnaces are fired to-day by powdered coal. The size of furnace varies from the small door type of only 18 by 18 inches, to the larger type suitable for cut lengths of billets up to 18 inches square section, having hearths of 12 by 6 feet.

#### **HEAVY FORGE FURNACES**

Blooms are heated by pulverised coal for serving both hammers and presses. The furnaces range from multiple small units dealing with 5-ton ingots, up to 50 and 100-ton ingots.

There are two main methods of applying powdered coal to forge furnaces; one is to cross fire into a side combustion chamber, allowing the gases to follow in their flow the contour of the roof, or alternatively the more direct method of applying the burners in the back wall of the combustion chamber and firing over the top of the ingot, care being taken that the direct flame from the burners does not impinge on the stock, particularly with alloy steels.

Care must also be taken to avoid the deposition of particles of fine ash on the metal. If this happens, the ash is forced into the metal under the hammer, and gives a faulty forging. This difficulty can be overcome to a large extent by installing, when space permits, a pre-combustion chamber before the furnace proper. The effect may also be minimised by ensuring consistently fine pul-

verisation of the fuel so that the ash particles are small enough to be carried over by the velocity of the gases.

## CONTINUOUS FURNACES

Powdered coal has been applied to many sizes and types of continuous furnaces in the steel industry, with outputs varying from  $\frac{1}{2}$  ton to 15 tons per hour. For ordinary mild steels the application is simple, two or more burners generally being placed in the back or combustion wall of the furnace and directed in a parallel line with the lie of the stock.

The actual design of furnace for specific purposes such as the production of solid drawn tubes, or other types of piercings must receive careful consideration, for in both of these applications the importance of the steel being really well soaked cannot be over-estimated. As an alternative to an increased rate of heat input at the discharging end, boosting burners can be placed half-way up the furnace.

In attaining uniformity of temperature much can be done by the design of the furnace as regards the slope of the furnace hearth and roof both of which have an effect on the manner in which the heat reaches the charging end of the furnace.

Alternatively, the furnace may be worked with a horizontal hearth with no slope and the temperature distribution attained by means of suitably placed multiple burners.

In the larger types of continuous furnaces for ingot heating dealing with alloy steels, modern practice is to fire both over and under the ingot and to have a soaking zone at the end of the furnace for producing uniformity of temperature.

## MELTING FURNACES

Powdered coal is used for melting iron for the production of malleable castings, and also in roll melting. The furnaces are of the reverberatory type, and the burners are generally placed in the back or combustion chamber wall of the furnace. Rotary or oscillating types are also applied.

## NON-FERROUS METALS

In the non-ferrous metal industry the main application is copper melting and refining. One of the earliest applications of pulverised fuel firing on a substantial scale was in the roasting of copper ores.

The use of pulverised coal has the following general features :—

- (1) Flexibility in the rate of combustion.
- (2) Close control of furnace atmosphere can be secured as a result of its having certain features in common with gaseous and liquid fuels.
- (3) A flame of high calorific intensity.
- (4) Special problems arise associated with the elimination of ash and grit carried in the flame and products of combustion.

## CHOICE OF FUEL

Generally speaking the moisture and ash content should be kept as low as possible, but here again the actual ash and moisture contents safely usable are governed entirely by the application. Thus, at collieries it is often quite sound practice to use de-dusted fines containing up to 30 per cent. of ash for the pulverised fuel firing of water-tube boilers. In this way the colliery can often find all its power requirements with a fuel which would otherwise be wasted. On the other hand to burn such a coal in say an ingot heating furnace would be impracticable due to clogging of flues and furnace and combustion chambers.

Most metallurgical applications call for a fuel of under 10 per cent. ash, whilst obviously the moisture should be kept as low as possible, though with

certain types of medium-speed mills moisture contents in excess of 10 per cent. can be dealt with.

On central plants the moisture should be kept very low, and should never exceed 6 to 8 per cent., or trouble can be anticipated in the pulverised coal bin itself.

*Fusion Point of Ash.* This is an important point; a governing factor is whether or not it is required to work with a dry bottom or dry furnace as for example in an annealing furnace, or alternatively whether the temperatures of the process are so high that a liquid bottom or slag tap operation is inevitable. With a slag tap a coal with a low fusion point of ash, which will facilitate the tapping of the furnace, is preferable.

Conversely, a high fusion point ash is required in the fuel with a dry furnace.

*Volatile Content.* The V.M. may vary between 20 and 40 per cent., but should not be lower than 20 per cent. Certain boiler applications can use volatile contents down to 5-10 per cent., but special arrangements have then to be made as regards secondary air distribution and, generally speaking, anthracite is not desirable for furnace work.

## FURNACE DESIGN

This is a specialised matter depending upon many considerations. The following are fundamental points of general interest.

*Burner Position.* The burner must be so positioned that the coal does not directly impinge on any refractory surface, otherwise there will be a very high rate of brickwork wear. More often than not, if there is a high rate of wear in a powdered coal furnace it is not due so much to excessive temperature as to the mechanical erosion of semi-burnt particles impinging on the walls. Here they continue to burn and combine chemically with the brickwork, causing rapid fluxing and erosion. A reducing action of the carbonaceous matter in the presence of ferruginous impurities in the ash can result in the formation of fluid magmas rich in ferrous silicate.

*Flue Design.* Flue design should always receive special attention. With pulverised coal the presence of ash is inevitable. Adequate provision for dealing with the ash is therefore imperative.

If the application calls for slagging in the furnace, an appreciable proportion will be dealt with in a liquid form and tapped out. If on the other hand the process is operated at temperatures below the range at which fluid slag is formed provision must be made for the removal of the ash from downtakes and flues.

Therefore the flue ports should, as far as possible, be large and few in number, and access should be provided for cleaning or rodding. It is often sound practice where room permits, to provide a small settling chamber between the furnace and the main flue. This is often possible in ingot heating furnaces and in some stamp furnaces.

The advantage is twofold; firstly this chamber acts as a settling chamber for the dust, and secondly, as such it acts as an extension chamber and lowers the temperature, and here again, with careful design, it is possible to ensure that the temperature drops from say, 1,100° C. or 1,200° C. to 800° C. The ash would then have dropped below nodulising temperature into the dry state where it can be effectively dealt with. There should be easy access to these chambers.

Another expedient to assist in ash removal is, assuming adequate draught to be available, to admit a small amount of cold air by means of an airbreak damper which will chill the particles so that they can be dealt with easily in a dry state.

Flues should be kept above ground for ease of access where possible, but

where due to site conditions they have to be below ground they should be of adequate proportions. Another sound point is to have access covers at such obvious blockage points as right angle bends.

The flues of continuous furnaces are generally a source of little trouble, as the exit gases should be at a low temperature, and therefore the question of nodulisation in the flue does not arise; the only problem is that of providing reasonable access.

*Air Recuperation.* The use of regenerators is not entirely impracticable, provided adequate means exist for the removal of accretions of dust and clinker.

A metal recuperator is recommended and if installed, the following points should be given close attention :—

(1) *Choice of Material.* An adequate grade of heat resisting steel or iron should be selected; amongst available materials are those capable of operating at temperatures up to 1,000° C. Alternatively, lower grade materials to withstand, say 600°–700° C., can be used, but the temperature of the incoming gases must be controlled accordingly.

(2) *Entry of Flue Gases to Recuperator.* It is sound to arrange for the temperature of the gases entering the recuperator chamber to be at least 80°–100° C. below the supposedly safe temperature of the metal selected. The gases can be adjusted to the required temperature level by by-pass flues or air infiltration. If air infiltration is allowed it is necessary to safeguard against the possibility of stratification of hot and cold streams with resultant distortion of the recuperator.

At a point just prior to the entry of the recuperator chamber a recording pyrometer with an electric connection can be provided, so that if the temperature rises above the selected point a warning horn can be sounded, or a red light shown, to attract the furnace attendant's attention.

(3) *Ash Accumulation.* The spacing of the metal elements should be fairly liberal, and the ash can be readily removed by means of dust blowers, following the normal practice in water-tube boilers, since the ash in the recuperator, due to the limit on maximum temperature of entry of the hot gases and the cooling effect of the tubes, will be in a dry dust form.

Another precaution, although it is not absolutely necessary provided the incoming temperature of the gases never reaches a dangerous limit, is so to arrange a fan that if hot recuperated secondary air is not being used on the furnace, a proportion is being blown to atmosphere by means of a bleeding-off pipe. This ensures that air will always pass through the recuperator.

## STORAGE OF COAL IN BINS

Under certain atmospheric conditions, particularly when there is considerable humidity and the coal is wet, there is a risk of the coal in storage bins heating up, and if this is allowed to continue indefinitely, spontaneous combustion will eventually begin.

As a rule there is no risk of explosion, but merely the heating up of the bin until the coal begins to glow. If this should occur, it can be dealt with by suitable CO<sub>2</sub> equipment.

The operation of the CO<sub>2</sub> supply can be made manually on the indication of pyrometers placed near the top as well as near the bottom of the bin, or alternatively the use of automatic fire prevention equipment controls the flooding of carbon dioxide into the bin.

## THE EFFICIENT USE OF PULVERISED FUEL IN BOILERS

The use of coal in the pulverised form as a fuel for boilers introduces many problems which are peculiar to this system of firing and, if the highest possible fuel economy is to be realised, these problems must be properly understood.

In this section attention is directed to the more important factors influencing the attainment of efficient operation to stimulate rational investigation by means of which the engineer in charge can find out for himself of what his plant is really capable and in what ways it is falling short of what is attainable.

In approaching the subject in this way from the point of view of an existing installation the first questions which must be answered are the following :—

- (1) What is the highest efficiency which it should be possible to obtain from the plant under service conditions ? This will be referred to throughout as the bogey efficiency, by analogy with golf.
- (2) In what respects is the operation of the plant falling short of this bogey efficiency and how can the inefficiencies be corrected ? Some of the inefficiencies will almost certainly be due to causes in no way associated with the use of pulverised coal and only those directly or indirectly due to this method of firing will be here examined in detail.

The best procedure to adopt in investigating the efficiency of working of a boiler plant is first to examine a comprehensive list of all factors on which the efficiency depends and to prepare a bogey energy balance sheet for the installation. In connection with each item should be considered any data available on design conditions, the present achieved operating conditions and the bogey efficiency conditions attainable consistent with limitations existing due to design or local conditions. Such a list is given later, but before considering in detail the problems which arise it will be well for the engineer in charge of the plant to satisfy himself that he is really clear as to the fundamentals of the steam-raising process and what effects will result from changes in the many variable conditions, for example the relation between  $\text{CO}_2$  in the flue gases and the overall thermal efficiency of his installation.

Whatever may be the class of fuel, the type of boiler and its operating steam pressure and temperature, or the purpose for which the steam is required, the basic principle is to convert economically the greatest possible amount of the energy in the fuel to energy in the steam leaving the boiler installation.

It is clearly impossible to realise a 100 per cent. conversion as some of the energy in the coal will, for example, be required to pump water into the boiler, some to drive air and gas fans and some to elevate and crush the coal. Losses must also occur from such factors as radiation from the boiler casing, the evaporation of moisture in the coal and heat carried to atmosphere in the chimney gases.

The above limitations can be set out in the form of an energy balance sheet : a typical sheet for a pulverised coal-fired boiler installation would be as in Table 62. This table gives bogey figures, as well as performance figures which might be obtained from a reasonably well operated installation, using a slack coal containing 15 per cent. ash and 3.5 per cent. moisture.

Table 62 does not take into account the energy required for coal crushing, screening and pulverising, for feed-water heating and pumping, and for such equipment as air and gas fans, etc. As the total energy for these purposes will not be inconsiderable, being of the order of 2 to 4 per cent. of the energy in the steam produced, every effort must be directed to obtaining the highest possible efficiency of operation of the auxiliary equipment.

Taking now the various items in the balance sheet (Table 62) consideration must be first given to the factors involved in each of these items and their relation to the magnitude of the loss, to the assessment of the "bogey" loss and to determining what steps can be taken to reduce the "actual performance loss." Brief mention only is made of these points where they are not of particular significance in pulverised coal firing, but whenever they are influenced largely by this method of firing sufficient detail is given to direct attention to the desirability and best method of investigation.

TABLE 62

Item	Percentage of energy in coal	
	Bogey	Actual performance
	Per cent.	Per cent.
1. Loss due to moisture in coal .. .. .	0.3	0.3
2. Loss due to moisture in air supplied to boiler ..	0.1	0.1
3. Loss due to moisture formed in burning hydrogen in coal .. .. .		3.6
4. Loss due to incomplete combustion of carbon in coal	0.1	0.2
5. Loss in dry gases discharged to atmosphere ..	3.4	4.6
6. Loss due to unburnt carbon in ash .. .. .	1.4	3.2
7. Loss due to radiation .. .. .	0.4	0.8
8. Loss due to heat losses in starting up and shutting down boilers .. .. .	0.1	0.2
9. Loss due to boiler water blow-down .. .. .	0.3	0.5
10. Loss due to steam for soot blowing .. .. .	0.3	0.5
11. Unaccounted for losses .. .. .	0.5	0.8
12. Energy in steam generated .. .. .	89.5	85.0
<b>TOTAL</b>	<b>100.0</b>	<b>100.0</b>

(1) *Loss due to Moisture in Coal.* All the moisture which is present with the coal as it is fed to the boilers will be converted to steam and this steam will be finally discharged to atmosphere at the temperature of the chimney gases with a consequent loss of heat energy. The loss of energy due to this cause is relatively small for normal moisture coals and the only ways in which the loss can be reduced are either by reduction of the moisture in the coal or a lowering of the temperature of the gases leaving the boiler system. The effect possible by the latter is extremely small and can be neglected.

Reduction of the moisture loss by reducing the moisture content of the coal can be brought about :—

- (a) By obtaining a coal with a lower moisture content.
- (b) By removing some of the moisture in a predrying plant in which heat energy which would otherwise be lost is used for the drying operation.

Unless the coal in use has a very high moisture content (b) above need not be considered. Added moisture, due to rain on unsheeted waggons or unprotected receiving hoppers, taken over a long period will have only a small effect on the fuel efficiency : it may, however, be an important factor in connection with the pulverising process : this aspect is dealt with later.

(2) and (3) *Losses due to Moisture in Air supplied to Boiler and to Moisture formed in Burning Hydrogen in Coal.* For all practical purposes both of these losses can be considered as fixed losses. The first, being of very small magnitude, is unimportant ; the second, which is quite considerable, is, however, quite unavoidable so far as concerns the latent heat of the resulting steam and most of its sensible heat.

(4) *Loss due to Incomplete Combustion of Carbon in Coal.* Unless operation is seriously at fault this loss will be very small with a well-designed installation. Abnormal conditions such as a sudden change in steam demand may lead to excessive coal feed to a burner or to incorrect adjustment of coal and air rates resulting in incomplete combustion of the coal, but, provided these conditions are infrequent, the effects on thermal efficiency will be very small.

Incomplete combustion of the carbon in the coal will be accompanied by a discharge of black smoke from the chimney, and boiler operators should be



encouraged to observe the condition of chimney gas at regular intervals if the position of the boiler operating floor permits of their doing this without interference with the necessary attention to the control of the boilers.

(5) *Loss in Dry Gases discharged to Atmosphere.* The magnitude of this loss, whilst, of course, being dependent primarily on the design of the installation in respect of such items of equipment as economisers and air preheaters, is, for any particular boiler, determined almost wholly by the quantity of air supplied to the furnace. This loss would be at a minimum value if it were possible to limit the air supply to the quantity theoretically necessary for complete combustion of the coal, but in practice about 30 per cent. excess air is necessary. The design of the boiler in respect of such items as type of burner, size, shape and degree of cooling of combustion chamber will determine the amount of excess air which must be supplied to ensure complete combustion of the coal, but as all these factors will be influenced by the fineness of the pulverised coal it is important that the optimum fineness should be determined.

Practical tests should be carried out to ascertain the degree of fineness of the pulverised coal necessary to ensure a minimum loss in the dry gases discharged to atmosphere. As, however, the loss due to unburnt carbon in ash (see (6) below) is also dependent very largely on the fineness of the pulverised coal the optimum operating condition must be determined from an examination of the following variables :—

- (a) Fineness of pulverised coal.
- (b) Energy consumed by pulverising equipment, air and gas fans, etc.
- (c) Loss in dry gases discharged to atmosphere.
- (d) Loss due to unburnt carbon in ash.

The nature of the necessary tests and the form of the relationship between the above variables is dealt with under (6).

(6) *Loss due to Unburnt Carbon in Ash.* In pulverised coal-fired installations this is the most important of the controllable losses and if the installation is not correctly operated may reach serious proportions.

When coal is pulverised the resultant powder is not merely a mixture of coal particles and ash particles, but of coal particles, ash particles and particles containing ash and coal intimately combined. When this powder is burnt in a boiler furnace the coal particles and a part only of the coal in the particles containing ash and coal are converted into gases; the remaining coal in the particles containing ash and coal is either deposited with these particles in the combustion chamber or is carried through the boiler system with them, entrained in the gases.

The portion that is deposited in the combustion chamber will vary in quantity as between one type of boiler and another, but seldom will be more than 15 to 25 per cent. of the whole; the coal in this fraction being subjected to furnace conditions for an appreciable length of time will, to a large extent, be burnt, but the remaining 75 to 85 per cent. of the particles will pass through the system unburnt due to the cooling effect of the boiler heat-absorbing surfaces.

The amount of coal in the ash particles which pass through the boiler system and which causes almost all the loss under consideration will depend on a number of factors, such as :—

- (a) The physical nature of the coal.
- (b) The ash content of the coal.
- (c) The fineness of the coal after pulverising.
- (d) The amount of excess air supplied to the combustion chamber.
- (e) Type of burner and combustion chamber.
- (f) The load on the boiler.

Except inasmuch as it may be possible to employ a coal containing less ash the only factors which demand consideration are (c) and (d), and an investigation of how to reduce the losses due to dry gases discharged to atmosphere and to unburnt carbon in ash should follow the lines set out below.

A series of tests, each preferably of not less than 24 hours' duration, should be made covering a fairly wide range of pulverised coal fineness and of excess air supplied to the furnace. Two tests at least should be made for each condition and the tests must be so arranged and conducted as to eliminate as far as possible any effects due to such factors as variation in type of coal, boiler load and cleanliness of the boiler system.

It is outside the present scope to discuss all the problems associated with such tests, but it cannot be emphasised too strongly that the greatest care must be taken to ensure that the results are not influenced by factors other than those under examination. As an example of the steps which should be taken to make certain of these conditions, the following is a procedure which is probably the best to ensure that the state of cleanliness of the boiler shall not affect the conclusions to be drawn from the tests.

Assuming that it has been decided to carry out first a series of tests to relate the unburnt carbon content of the ash with the fineness of the pulverised coal, the first set of tests should be started with a boiler which has just been cleaned and should be worked up through a range of fineness from the finest to the coarsest grading to be tested. On completion of this set of tests the boiler should be cleaned and a second set of tests made, working down from the coarsest to the finest gradings.

As an example of the results which might be expected from such tests results are here given obtained from an installation using a coal having the following analysis :—

Ash .. .. .	15.0 per cent.
Moisture .. .. .	3.5 "
Carbon .. .. .	68.0 "
Hydrogen .. .. .	4.5 "
Sulphur .. .. .	3.0 "
Oxygen and nitrogen .. .. .	6.0 "

TABLE 63. RELATION BETWEEN COAL FINENESS AND CARBON IN ASH

Pulverised coal fineness :					
Percentage > 100 I.M.M. mesh	3.1	5.7	9.8	13.1	19.3
Average carbon in ash .. .. .	10.1	12.1	17.4	20.7	

The results in Table 63 were for the excess air rate to the boiler found from previous tests to be the most economical. From these and other data obtained during the tests the results in Table 64 were obtained; an analysis of these

TABLE

Item	Units					
Fineness of pulverising	Percentage > 100 I.M.M. mesh	2	5	8	11	14
Carbon in flue dust ..	Percentage ..	9.1	11.7	14.9	18.3	21.9
Boiler thermal efficiency ..	Percentage ..	86.75	86.30	85.72	84.92	83.98
Mill output .. .. .	Tons/hour/mill ..	8.1	9.2	10.0	10.6	11.2
Mill power consumption ..	kWh/ton of coal	30.1	27.7	25.6	24.0	23.1

results showed that the economic fineness of pulverised coal for the installation was 4 per cent.  $> 100$  I.M.M. mesh.

It will be noted that the power consumption of the mill is very considerable; this was due to the use of a coal with a poor grindability and the employment of ball mills operating on the storage system. The power consumption of the mill air fans is included.

*Losses due to (7) Radiation, (8) Starting Up and Shutting Down Boilers, and (9) Boiler Water Blow-down.* The use of pulverised coal as a fuel has no special effect on the magnitude of these losses which, in any event, being small in properly maintained installations, are not potential sources of any considerable fuel savings. It is, nevertheless, important to ensure that the lagging on the system is both adequate and kept in good condition, that unnecessary boiler shut downs are avoided and that the losses due to boiler water blow-down are minimised by the adoption, if possible, of a continuous blow-down system with heat recovery from flash-off steam.

*(10) Loss due to Steam for Soot Blowing.* With pulverised coal firing the quantity of ash carried forward in the boiler system is always very considerable except where very low-ash coals are used, and in many installations may amount to as much as 0.75 to 1.0 ton for every 100,000 lb. of steam generated. If this ash is allowed to accumulate on the boiler, economiser and air preheater surfaces the heat absorption of the system will be impaired. Removal of these deposits by soot blowers is, therefore, necessary, but it must not be forgotten that soot blowers, whether they operate with steam or compressed air, absorb energy, and that it is important that their positioning, as well as the sequence and frequency of their operation, should be the best possible.

It is difficult to carry out tests on soot blowing, but if the soot blowers are always operated to a strict routine with a timed period of operation of each soot blower the effect of precise changes in the routine can be observed.

It is probably true to say that in the majority of installations the soot blowers are used excessively with a resultant lowering of the overall efficiency.

*(11) Unaccounted for Losses.* It may be considered strange that in Table 62 the bogey value of these losses is not given as zero. The reason for this is that under this heading are included items which are not losses in the strict sense of the word. This item in the energy balance covers really the errors which exist in the other items in the balance sheet due to inaccuracies in metering steam, recording  $\text{CO}_2$  in gases, measuring gas temperatures, and so on.

Any great variations in the value of this item in the routine plant balance sheets must be taken as an indication that instruments are not functioning properly or are incorrectly read, or that there has been faulty operation of, for example, soot blowers. The energy balance loss in the case of soot blowers is usually based, not on metered steam quantities, but on quantities calculated for a defined routine operation of the blowers.

Before leaving the subject of the energy balance sheet for the installation it will be well to recapitulate the issues which are of importance in a pulverised fuel fired installation.

The energy losses which are particularly affected by the use of the pulverised coal system of firing as compared with the grate system are :—

- (a) Loss in dry gases discharged to atmosphere.
- (b) Loss due to unburnt carbon in ash.

The first of these losses should be lower with pulverised coal than with grate firing since with pulverised fuel it is possible to use a smaller quantity of excess air and by the use of higher air preheat, amongst other factors, to attain a lower exit gas temperature and, therefore, a smaller loss due to dry flue gases.

With pulverised fuel firing the loss due to unburnt carbon in ash will be almost invariably greater than with grate-firing and unless the matter has been thoroughly investigated this loss will, in the vast majority of plants, be one which can be appreciably reduced.

The point which must, therefore, be stressed is that the precise magnitude of these losses under existing operating conditions should first be determined, and, from data obtained from carefully conducted and scientifically planned tests, the optimum practical operating conditions should be ascertained.

In Table 65 are set out typical balance sheets for pulverised fuel and for chain-grate fired boilers, for the type of coal referred to earlier in this chapter. The higher loss figures under columns "A" and "C" are those which might be occurring in a well-designed plant indifferently operated; the figures under columns "B" and "D" represent conditions closely approaching the bogey losses for the installations.

65

Item	Percentage of energy in coal			
	Pulverised fuel		Chain-grate stoker	
	A	B	C	D
1. Loss due to moisture in coal .. .. .	0.3	0.3	1.5	1.4
2. Loss due to moisture in air supplied to boiler ..	0.1	0.1	0.1	0.1
3. Loss due to moisture formed in burning hydrogen in coal .. .. .	3.6	3.6	3.7	3.7
4. Loss due to incomplete combustion of carbon in coal ..	0.2	0.1	0.3	0.1
5. Loss in dry gases discharged to atmosphere ..	4.6	3.4	7.6	5.7
6. Loss due to unburnt carbon in ash .. .. .	4.0	1.8	3.1	0.7
7. Loss due to radiation .. .. .	0.8	0.5	0.8	0.5
8. Loss due to heat losses in starting up and shutting down boilers .. .. .	0.2	0.1	0.2	0.1
9. Loss due to boiler water blow-down .. .. .	0.5	0.3	0.5	0.3
10. Loss due to steam for soot blowing .. .. .	0.5	0.3	0.5	0.3
11. Unaccounted for losses .. .. .	0.8	0.5	0.8	0.5
Total losses .. .. .	15.6	11.0	19.1	13.4
Gross overall thermal efficiency .. .. .	84.4	89.0	80.9	86.6

Consideration has now been given to all the principal factors which are associated with the combustion process and it remains to deal with the general survey of the installation mentioned in the opening paragraphs of this section.

To present a really comprehensive survey of all the items of equipment in a boiler installation is outside the present scope. To indicate, however, the method of examination which is recommended a partial survey of a typical boiler plant is set out in Table 66. The points covered by this survey are by no means all which will have to be considered, but should be sufficient to indicate the method of approach to the problem.

If a survey on these lines is conscientiously carried out it is almost certain to draw attention to items about which the engineer in charge lacks complete information or in connection with which improved conditions are possible. There is very little doubt that in all installations some, if only a small, improvement in overall efficiency is possible and that in many installations very substantial savings can be effected.

Efficient operation of boiler installations is of vital importance and no time should be lost or effort spared in effecting all economically practicable improvements. The boiler installation is too often considered as of importance only

in so much as it shall continue to produce the quantity of steam required ; much is lost by this attitude and if engineers would devote more time to a scientific investigation of the performance of their installations they would make a very substantial contribution to the national effort, and would, at the same time, find in their investigations much of real interest to them as engineers.

The best possible advice to the engineers in charge can be summed up in a few sentences :—

Know your installation in the greatest possible detail ; its capabilities and its shortcomings.

Take a very close interest in the operators and maintenance personnel and make sure that they, too, understand thoroughly the portions of the installation for which they are responsible.

Study the performance of the installation and by properly conducted tests ascertain the optimum conditions of operation.

Make sure that the information on which the installation is controlled, for example, the fineness of the pulverised coal, the  $\text{CO}_2$  in the flue gases, the unburnt carbon in the ash, is reliable and is properly used.

Provide the operating and maintenance staff with clear and precise instructions and make sure that they understand and carry them out.

TABLE 66. SURVEY OF PULVERISED COAL-FIRED BOILER INSTALLATION  
(Storage System)

Item	Particulars of equipment and other information	Matters to be considered														
<i>Raw coal</i>	<p>Coal consumption of plant, 2,600 tons/week. Coal from three sources in varying quantities. Average analysis of all coal for past twelve months.</p> <table><thead><tr><th></th><th>Per cent.</th></tr></thead><tbody><tr><td>Ash .. .. .</td><td>15.0</td></tr><tr><td>Moisture .. ..</td><td>3.5</td></tr><tr><td>Carbon .. .. .</td><td>68.0</td></tr><tr><td>Hydrogen .. ..</td><td>4.5</td></tr><tr><td>Sulphur .. .. .</td><td>3.0</td></tr><tr><td>Oxygen and nitrogen</td><td>6.0</td></tr></tbody></table> <p>Moisture content at times as high as 5.5 per cent.</p>		Per cent.	Ash .. .. .	15.0	Moisture .. ..	3.5	Carbon .. .. .	68.0	Hydrogen .. ..	4.5	Sulphur .. .. .	3.0	Oxygen and nitrogen	6.0	<p>Can a lower ash coal be obtained in order to reduce unburnt carbon in ash losses, mill wear and power consumption, and rate of slag formation in furnace ?</p> <p>As choking and corrosion of air pre-heaters is being experienced could a lower sulphur content coal be obtained or could arrangements be made on the mills to purge out some of the pyrites in the coal ?</p>
	Per cent.															
Ash .. .. .	15.0															
Moisture .. ..	3.5															
Carbon .. .. .	68.0															
Hydrogen .. ..	4.5															
Sulphur .. .. .	3.0															
Oxygen and nitrogen	6.0															
<i>Raw coal handling</i>	<p>All coal is rail-borne direct to plant. 2 receiving hoppers at ground level (not roofed). 2 skip hoist elevators. 2 band conveyors.</p>	<p>Are waggons sheeted and if so, are sheets removed only immediately before discharging the waggons in rainy weather ?</p> <p>Are mill operators informed by off-loading operators of the type and condition of the coal which is being elevated and the bunkers to which it is being fed ?</p> <p>Are the loading valves correctly adjusted to ensure a full skip bucket and so keep operating time and, therefore, power consumption of elevators to a minimum ?</p> <p>Is there any spillage of coal from the skips, and if so, is the coal being collected and returned to the hoppers ?</p> <p>Is the no-load power consumption of the band conveyors satisfactory ? Are the conveyor idlers properly lubricated and are steps taken to ensure that the free motion of the return half of the band is not restricted by spillage coal ?</p> <p>Are conveyors stopped immediately the coal supply to them has ceased ?</p>														

Item	Particulars of equipment and other information	Matters to be considered
<i>Pulverising equipment</i>	<p>4 ball mills with circulating air fans and cyclones. 5 tons/hour rated capacity with coal of type referred to under "Raw Coal" above and with grading of pulverised coal 8 per cent. &gt; 100 I.M.M. mesh. Actual performance—4.6 tons/hour at 6 to 8 per cent. &gt; 100 I.M.M. mesh. Coal fed to mills by variable speed rotary feeders. Mill systems supplied with cold air from boiler house and vented to boiler primary air fans. Atmospheric vents provided for starting up and abnormal conditions.</p>	<p>When a batch of wet coal is received, is this coal distributed over a number of bunkers or otherwise mixed with drier coal to minimise effect on the mills?</p> <p>In view of performance of mills being below specification what information is available on the following points:—</p> <ol style="list-style-type: none"> <li>What is the condition of the mill linings?</li> <li>What is the condition and weight of the ball charges? Has the optimum ball charge been determined by plant tests? What is the procedure for maintaining the ball charge? What is the ball wear per ton of coal pulverised?</li> <li>Is the correct circulating air rate being maintained?</li> <li>Are the grading tests of the pulverised coal systematic and reliable? What is the condition of the test sieves and are they checked against a standard sieve?</li> <li>Does the performance vary as between one mill and another and with one operator and another?</li> <li>What is the economic fineness of pulverised coal for the installation?</li> <li>Is coal being wasted by improper use of the mill system atmospheric vent or by continuous leakage through the vent valves?</li> <li>Is sufficient interest being taken by the engineers in charge in the performance of the pulverising equipment and has the significance of maintaining the correct final product fineness been properly explained to the operators?</li> </ol>
<i>Firing equipment</i>	<p>4 spider feeders per boiler each feeding coal from the pulverised coal bunker to the primary air stream to the four burners. 4 turbulent type burners with refractory burner mouths set in refractory front wall of combustion chamber. Trouble experienced fairly frequently due to spider feeders tripping out due to choking with coal. Trouble experienced with slag formation on burner mouths. This slag usually on lower part of mouth.</p>	<p>Do the feeders maintain at all speeds a steady coal supply to the burners? Do the feeders, possibly as a result of wrong adjustment or of wear, give rise to aggregation of the coal particles with consequent losses due to incomplete combustion? Is this a cause of the slag formation on the burner mouths? Has the correct ratio of primary to secondary air been determined by plant tests? Are the burner air louvres correctly adjusted? When refractory repairs are carried out is proper care taken to ensure symmetry and correct contouring of the burner mouths? Are adequate instruments available to enable the operator to control the boiler efficiently? Are the instruments properly maintained so that complete reliance can be placed on them?</p>

Item	Particulars of equipment and other information	Matters to be considered
		<p>Is the CO<sub>2</sub> recorder sampling connection positioned so that a true sample of the flue gases is obtained or is a false reading registered due to stratification of the gases in the boiler flues ?</p> <p>Are all boiler instrument charts and record sheets examined by a competent engineer daily, and is any action, indicated by them as being necessary, taken immediately ?</p>
<i>Combustion chamber</i>	<p>Rating : 2·0-lb. coal/ft.<sup>2</sup>/hr. at design load.</p>	<p>Are combustion chamber soot blowers operated at sufficiently frequent intervals to ensure that deposited ash particles will not sinter and form slag accumulations which cannot be removed by soot blowers ?</p> <p>Is the combustion chamber provided with adequate and suitably placed inspection openings and do operators keep a sufficiently close watch on combustion chamber conditions ? As the combustion chamber rating is high, is care taken to keep individual boiler loads as low as possible by equal distribution of the load ?</p>
<i>Boiler heating surface and flue gas system</i>	<p>Tri-drum boiler with single gas pass: radiant heat and convection superheaters ; steaming economiser and regenerative air preheater.</p> <p>Trouble experienced with slag formation on front row of boiler tubes. The rate of formation varies considerably as between one boiler and another. Considerable trouble experienced with No. 4 boiler air preheater with dust deposits and corrosion of plates.</p>	<p>Is the soot-blowing routine properly planned, efficiently carried out and recorded on the boiler record sheets ?</p> <p>Is care taken to ensure that in extremely cold weather the temperature of the air to the air preheater is not allowed to become so low that trouble results from flue dust adhesion to the heating surfaces at the cold end ?</p> <p>Is steam from superheater drains, steam traps, ash sluice channels, etc., allowed to enter the forced draught fan suction resulting in troubles in the air preheaters due to high moisture content of air ?</p>
<i>Boiler fittings and steam and water systems</i>	<p>Trouble has been experienced with safety valves not shutting off completely after blowing.</p>	<p>Are the safety valves absolutely steam-tight under normal working pressure conditions ?</p> <p>Are the water level gauges positioned at the right level and is the water level maintained at such a level as to ensure that carry-over of water with the steam is kept at a minimum quantity ?</p> <p>Is the boiler water blow-down procedure controlled in such a way that whilst maintaining correct boiler water conditions the heat losses are kept at a minimum value ?</p> <p>Is the steam flashed from the blow-down water used by delivering it into the low pressure mains for heating or other purposes ?</p> <p>Are steam trap lines effectively lagged so that the discharge of the trap is not due more to condensation in the trap lines than in the steam mains drained by the traps ?</p>

## CHAPTER XIII

### STEAM BOILERS

Boiler design—Vertical boilers—Stationary locomotive boilers—Cornish boilers—Lancashire boilers—Externally-fired boilers—Economic boilers—Waste heat boilers—Water-tube boilers—Effect of prime mover on boiler design—Feed water temperature—Furnace design—Water circulation.

**W**ITHIN the limits of modern knowledge, steam forms the most convenient and the most economical vehicle of heat for power and process purposes. Other substances, such as mercury and diphenyl, have been employed, but none of these is so universally available as the water from which steam is generated. Steam boilers therefore predominate in industry, and range in size from the elementary unit evaporating a few gallons per hour to the complicated power-station steam generator with an output of many thousand pounds per hour.

The purpose of a steam boiler, or more correctly a steam generator, is to produce steam under pressure from the raw materials fuel, water and air. The potential heat of the fuel is set free and this available heat then transmitted to, and stored by, water vapour in the form of sensible and latent heat.

The change and transference should be made with the minimum loss of heat, i.e. at maximum efficiency, it always being borne in mind that so far there have been financial, as well as practical, limitations to the degree of efficiency, but that in view of the urgent need to conserve fuel, the financial limitations may have to be somewhat relaxed.

Given equally efficient firing conditions, therefore, a good average thermal efficiency must be inherent in a boiler design by virtue of the disposition and form of its heating surface.

With some designs or types, the highest efficiencies can only be obtained by adding accessories to the boiler such as superheaters, air heaters, economisers, etc., but in other types such additions would lead to practical difficulties in the operation and excessive maintenance and repairs.

#### THE EMPIRICAL NATURE OF BOILER DESIGN

The development of steam boilers has so far proceeded along the lines of trial and error rather than by deliberate design based on theoretical consideration, since our knowledge has been insufficient to admit of any other course. To-day, however, accumulated knowledge enables the true boiler designer to design with somewhat more confidence and to forecast with reasonable accuracy the general conditions that may be expected to prevail within the boiler.

As the outcome of experience, boiler makers in general have now reduced the number of types to about six or seven, one or other of which have been found to be best suited to certain specific conditions. Thus the vertical cross-tube boiler is installed in factories requiring small amounts of steam, an improved type being employed when a greater output and higher economy is desired.

The stationary locomotive boiler is favoured by some trades because of its fairly rapid steam-raising qualities, and others prefer the larger Cornish or Lancashire types on account of their greater reserve and ability to satisfy fluctuating steam demands.

When space is limited and outputs of 1,000 to 30,000 lb. of steam per boiler per hour are required, the Economic boiler is widely used. In large capacity industrial and power-station installations where high working pressures are required, the water-tube boiler is necessarily adopted. Transport is one of the limiting factors in the manufacture of shell boilers.



## EVAPORATIVE CAPACITY AND GRATE AREA

In all these widely varying types of steam boiler, the evaporative capacity is a function of the amount of fuel that can be burned in the firebox or furnace. It is also a function of the calorific value of the fuel used, since obviously a grate designed for a combustible such as wood fuel must be larger than one using a good quality coal, if the same heat input is desired.

In a vertical cross-tube boiler similar to that illustrated in Fig. 94, the diameter of the boiler is directly related to the area of grate required to burn a given amount of fuel. Boilers of this class are normally coal-fired under natural draught, and can burn from 14 to 20 lb. of coal per hour per square foot of grate area. Again, 1 lb. of coal burned in one of these boilers will

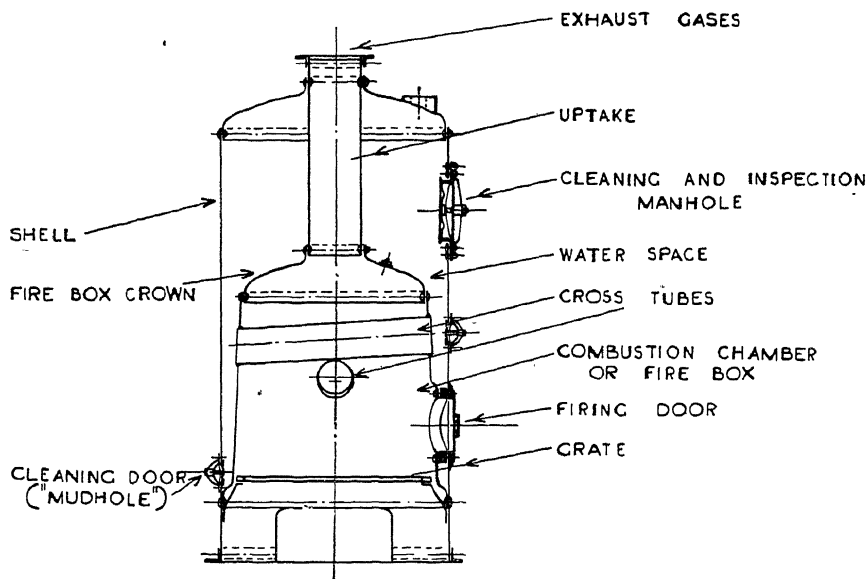


FIG. 94. Vertical cross-tube boiler.

evaporate from 5 to 8 lb. of water from and at 212° F., so that the computation of fire-grate area for a given evaporation is a comparatively simple matter.

## VERTICAL CROSS-TUBE BOILER PROPORTIONS

As an example—and the method applies equally to all internally fired boilers, with suitable modifications—there is here considered a vertical cross-tube boiler required to produce about 1,700 lb. of steam per hour (from and at 212° F.). It is assumed that 18 lb. of coal are burned per square foot of grate area, and that each lb. of coal will evaporate 6 lb. of water.

The evaporation per square foot of grate area will be  $6 \times 18 = 108$  lb., and the grate area required will be  $1,700/108 = 15.75$ , say, 15.8 square feet. This gives a grate diameter of 54 inches, and, allowing, say,  $\frac{1}{2}$ -inch for the fire-box plate thickness, the external diameter of the firebox is 55 inches. A space of at least  $2\frac{1}{2}$  inches must be allowed between the firebox and shell for cleaning and circulation, so that the shell diameter must be 60 inches.

The number of cross-tubes fitted varies from one to six, being arbitrarily determined according to the boiler diameter. In the boiler under considera-

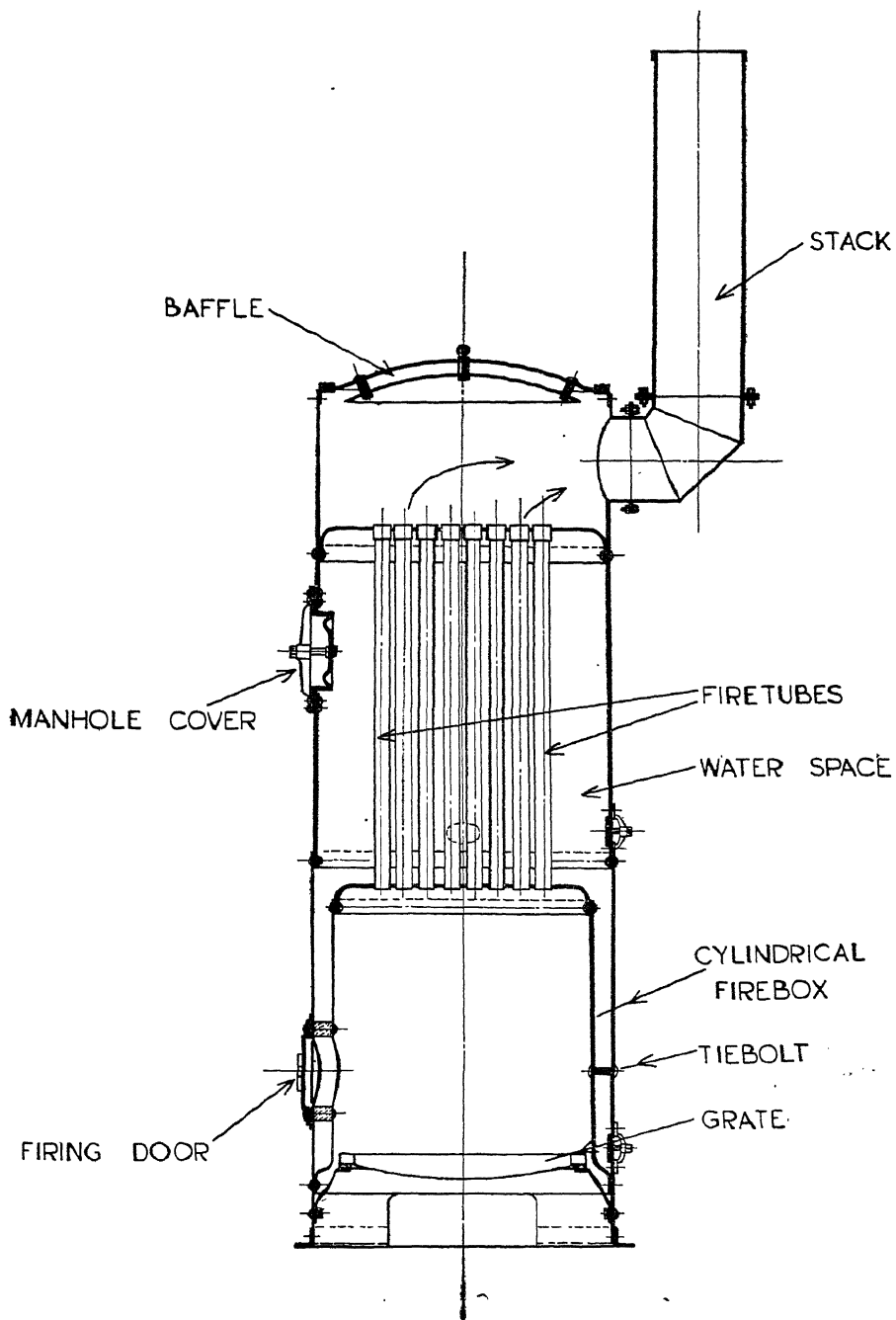


FIG. 95. Vertical fire-tube boiler with vertical tubes.

tion there would be about five, each 9 inches internal diameter, and to accommodate these and allow for the firehole and 10-12 inches of water leg below the firehole, an overall firebox height (measured to the top of the crown) of about 7 feet would result.

Similarly, the uptake diameter is usually about one-quarter of the shell diameter; most makers have standard sizes. In the present example an uptake having an internal diameter of 14 inches would suffice.

The distance between the top of the firebox crown and that of the shell crown depends upon the steam storage space allowed. The steam user has little choice in this respect when purchasing a standard boiler, but average allowances range from 1.75 to 3 cubic feet per 100 lb. of rated capacity.

### HEAT TRANSMISSION IN VERTICAL CROSS-TUBE BOILERS

An examination of Fig. 94 will show that the bulk of the heat transmission in a vertical cross-tube boiler must be effected by radiation. The gases from the furnace pass upwards to the chimney via segmental passages formed by the sides of the firebox and the cross-tubes and these passages are large in proportion to the volume of the gases passing. The result is that turbulence is slight or non-existent, and much gas does not get an opportunity of contact with the heating surface in order to give up its heat and as a result the outlet temperature of the gases is high and chimney losses also high with consequent low thermal efficiency, often as low as 50 per cent. and sometimes even lower.

Boilers of this class have the important advantage that they occupy little floor space and do not need elaborate foundations. Their popularity in this country may be attributed to these characteristics and to their ability to operate with considerable latitude in regard to the quality of feed water and fuel used. They are used, for example, as crane boilers. They should be lagged and a regulating damper should be provided in the chimney, operated from ground level.

### VERTICAL SMOKE-TUBE BOILERS

The vertical cross-tube boiler, as stated above, is not an efficient steam generator, and when outputs greater than 1,500 lb. per hour are required it is advisable to consider the use of a boiler having a greater proportion of tubular convective heating surface. This is to be found in the vertical smoke-tube boiler which is built as two main types.

Fig. 95 shows a smoke-tube boiler with vertical tubes, and this is, in effect, a plain vertical boiler with a shortened cylindrical firebox, the crown of which is perforated to take the tubes. The firebox height varies from 0.75 to 1 diameter to ensure good combustion, and the sides and crown absorb the radiant heat of the burning fuel.

The products of combustion, instead of having an almost unrestricted flow out of the boiler as in the vertical cross-tube type, are split up into a number of small gas streams which give up their heat to the tube surface in passing through the water and steam spaces of the boiler. Increased efficiency results, and as a rule the vertical smoke-tube boiler has a slightly greater evaporative capacity than a cross-tube boiler of the same overall dimensions.

With this type of boiler there is generally a good deal of corrosion of the tubes about the water line, and where the water is not good there is a liability of trouble at the furnace crown owing to deposition of scale.

Fig. 96 shows the other basic type of vertical boiler with horizontal smoke tubes, and it will be noted that the hemispherical firebox is admirably constructed for radiant heat absorption and that considerable area in the form of small diameter tubes is provided for the absorption of heat by convection. It has the additional important operating advantage that no seams or tube ends are exposed to the direct heat of the fire.

Further, the refractory-lined chamber arranged in way of the back tube plate serves to complete the combustion process.

The efficiency is generally of a reasonably high order and the boiler can be used for capacities up to 6,000 lb. of steam per hour under easy steaming conditions.

Another design of this type has an entirely submerged combustion chamber, thus giving more radiant heat absorbing surface.

The ratio of heating surface to grate area in the form of smoke-tube boiler just described varies from 10 to 26 : 1, against the cross-tube boiler ratio of about 8 to 10 : 1, and higher combustion rates up to 25 lb. of coal per square foot of grate area are permissible.

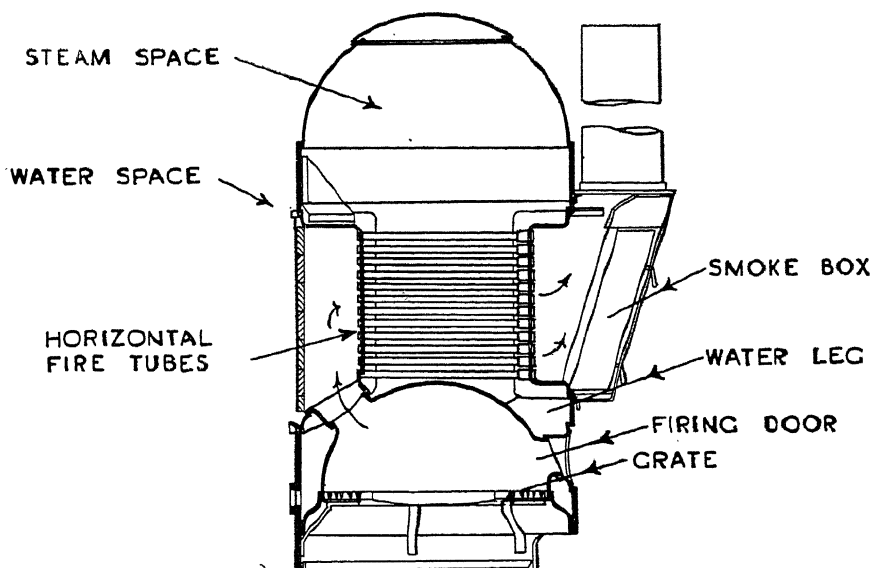


FIG. 96. Vertical fire-tube boiler with horizontal tubes.

### HEAT TRANSMISSION IN SMOKE- AND WATER-TUBES

In Chapter VIII the adverse effect of gas films on heat transmission was discussed. In a smoke-tube the amount of heat given up by a hot gas flowing through it depends upon the number of molecules coming into contact with the metallic wall. If the tube is too short, or its diameter is too large, the heat-resisting film of gas molecules adjacent to the wall remains more or less undisturbed, and since its thermal conductivity is about one-fifth of that of a good-quality insulating brick, heat transmission is impaired in consequence.

On the other hand, if water-tubes are suitably disposed in a gas flow, instead of a corresponding nest of smoke-tubes, there is an increased tendency for the gases to "scrub" the external surfaces, so continuously removing the gas film and benefiting heat transmission.

Experiments indicate that a water-tube nest of this type may be up to 80 per cent. more efficient than the equivalent number of smoke-tubes, and, when this principle is applied to the ordinary vertical boiler, it has the important result that the water circulation is increased in the restricted water leg between firebox and shell.

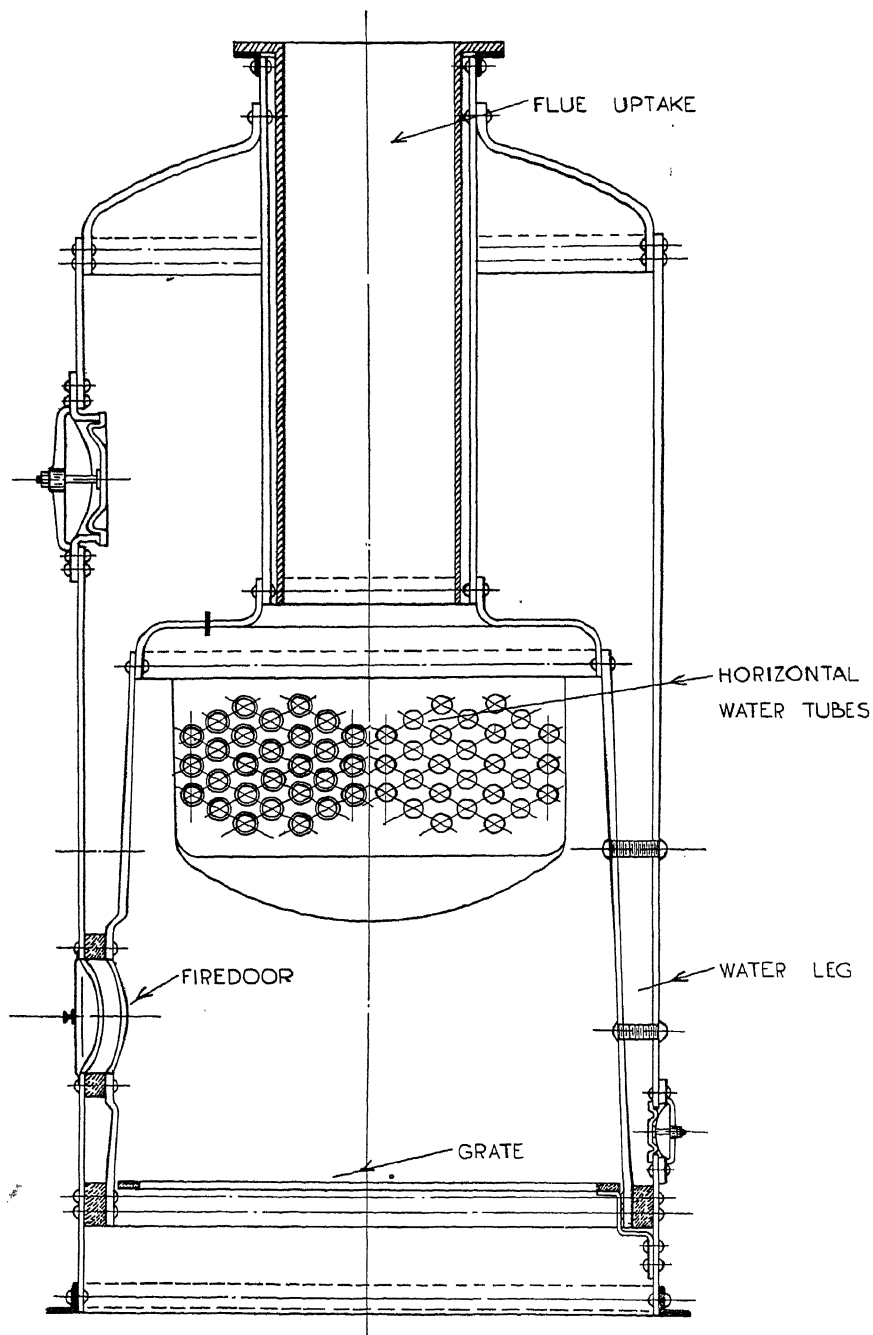


FIG. 97. Vertical water-tube boiler with horizontal tubes.

## VERTICAL WATER-TUBE BOILERS

A basic type of vertical boiler with horizontal water-tubes is shown in Fig. 97, the cylindrical vertical firebox being modified to form tube plates for the water-tubes. The lower parts of the firebox and the lower tube rows are subjected to the radiant heat of the fire, while the staggered nest of tubes in the upper part of the firebox is effectively scrubbed during the passage of the gases to the conventional type of uptake.

The water-tubes are given a slight inclination to the horizontal, causing a positive and uni-directional flow of water across the boiler. Steam-raising, in consequence, is fairly rapid, and this type of boiler is obtainable for outputs up to 6,000 lb. per hour. A thermal efficiency of about 70 per cent. can be obtained under reasonably good operating conditions with a well-designed boiler of this type.

## STATIONARY LOCOMOTIVE BOILERS

From the foregoing remarks it will be appreciated that one of the most difficult problems facing the vertical boiler designer is the production of a boiler at a commercially suitable price, while providing sufficient convective heating surface to absorb the residual furnace heat resulting from the high rate of combustion necessary in the firebox to keep down the overall dimensions (and cost) of the boiler.

This is a form of vicious circle, and some steam users have turned to the stationary locomotive type of boiler as a solution of the problem when requiring outputs between about 2,000 and 5,000 lb. per hour; the locomotive boiler of less than 2,000 lb. per hour capacity is rather restricted in cleaning and inspection facilities internally if the barrel is much under 42 inches in diameter. Boilers of this type have been made for producing 9,000 lb. of steam per hour.

Like the vertical boiler, it does not require any elaborate foundations, and it is practically self-contained as shown in Fig. 98.

Its other advantages are that it can be designed to incorporate a comparatively large area of heating surface per unit volume, it is a fairly rapid steam generator and it will stand a limited amount of forcing.

The cubical firebox is water cooled at its sides and top, a construction which allows high heat-release figures per cubic foot, but has the disadvantage of poor water circulation in the surrounding vertical water "legs."

The horizontal dimensions of the firebox are determined from the weight of coal that it is possible to burn per square foot of grate area, but the height is a compromise between the requirements of adequate combustion space, the area required for the tube plate (which, of course, also affects the barrel diameter and cost), and the steam space required above the working water level if priming is to be avoided when steaming.

Combustion rates vary from 16 to 25 lb. of coal per square foot of grate area under natural draught, and from 6 to 9 lb. of water can be evaporated per lb. of coal burned. These rates are, of course, very much less than those obtained in railway practice, where the rate is sometimes as high as 100 lb. per square foot as a result of the induced draught effect obtained from the engine exhaust and the better circulation and movement of the water resulting from vibrations. Stationary locomotive boilers having an evaporative capacity of 4,000 lb. per hour, for example, would require a grate area of about 24 square feet, under natural draught conditions of operation.

The firebox height, as has been already mentioned, is limited in a boiler of this type because of its encroachment on the steam space, and also to a lesser extent, because of expansion arising in service from the use of high fireboxes. A cubical firebox is not the ideal form for withstanding high temperatures and pressures, and undue forcing may result in strained seams and tube-plate

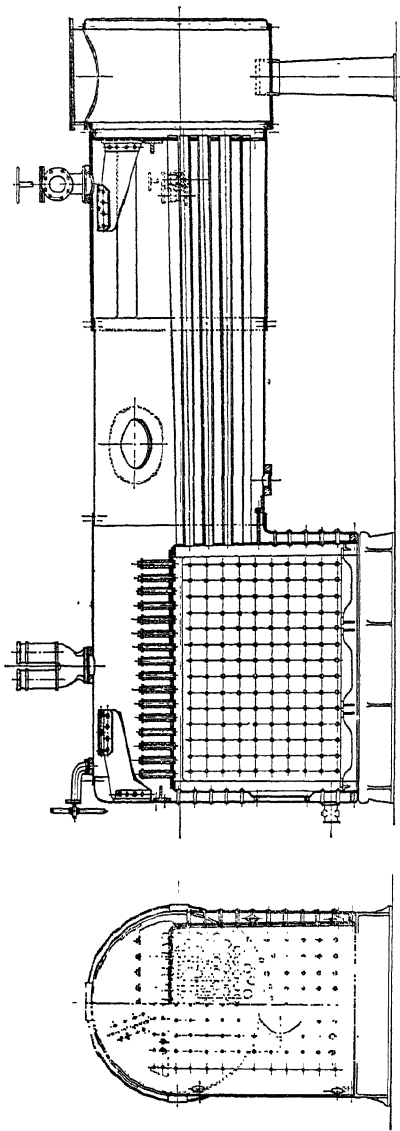


FIG. 98. Stationary locomotive boiler.

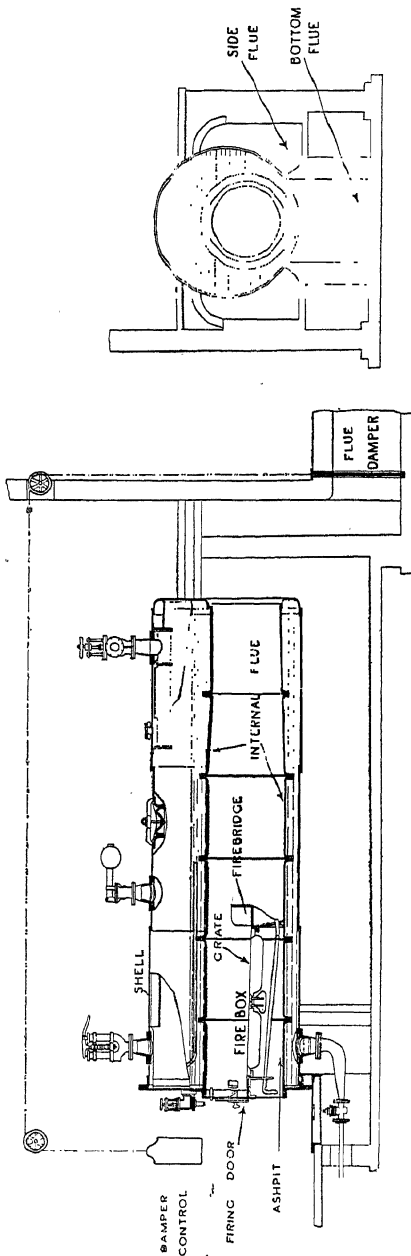


FIG. 99. Cornish boiler.

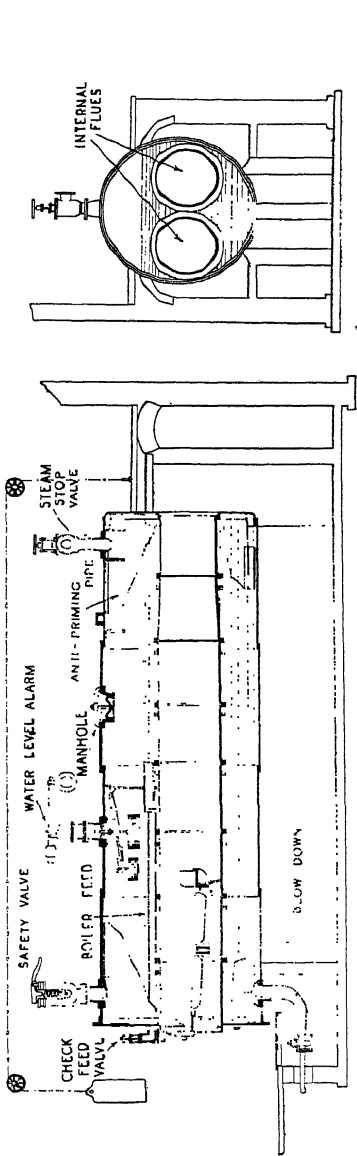


Fig. 100. Lancashire boiler.

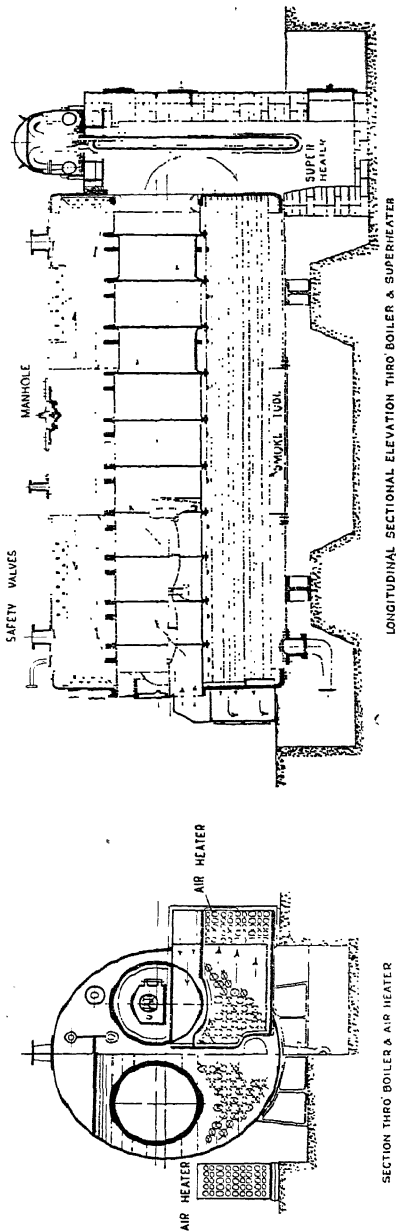


Fig. 101. Modified Lancashire boiler.



troubles. In consequence, the thickness of steel fireboxes is restricted to a maximum of about  $\frac{3}{16}$  inch and copper fireboxes to about  $\frac{1}{8}$  inch, and this, with the minimum stay pitch necessary for flexibility plus strength, restricts the permissible working pressure to a maximum of 300 lb. per square inch.

Thus the maximum area of boiler heating surface exposed to radiant heat is curtailed by constructional features. The convective tubular heating surface is similarly limited in regard to cross-sectional area, the last mentioned being equal to about one-sixth of the grate area, but more scope is permissible in regard to length, within the requirements of a reasonable total manufacturing cost and draught loss. Obviously, a long tube allows more opportunity for heat transmission between gas and tube wall than a short one, and a small-bore tube gives better heat transference than does one having a larger cross-sectional area. Long tubes, however, may set up uneconomical draught losses, and small-bore tubes are liable to become choked with soot.

The length of stationary locomotive boiler tubes in British practice is therefore between 30 and 50 diameters, and the minimum is  $2\frac{1}{4}$  inches (external). The ratio of total heating surface to grate area varies from 15 : 1 in the smallest to 40 : 1 in the largest boilers.

A disadvantage of the locomotive boiler is that the water circulation is restricted over those areas subjected to the highest temperature difference, so that overheating at some time or another is almost certain unless a fairly pure feed water is used.

Similarly, a boiler of this type designed and manufactured for cheapness will give poor results, because, to reduce the cost of material, the boiler-maker assembles the maximum tubular heating surface into the minimum barrel diameter. This limits the free flow of water essential to good circulation. A study of the best practice shows that a space of about  $\frac{3}{4}$  inch is left around each tube for circulation purposes, and the distance between the periphery of the outer tubes and the barrel is rarely less than  $2\frac{1}{2}$ -3 inches.

Boilers of the Cornish and Lancashire class, on the other hand, do not suffer from disadvantages of this nature, and, because of the amount of abuse they will successfully withstand in these respects, they are widely used in factories requiring up to about 12,000 lb. of steam per hour per boiler at pressures up to 250 lb. per square inch.

## CORNISH AND LANCASHIRE BOILERS

The Cornish or single internal-flue "shell" boiler shown in Fig. 99 is made for evaporative capacities ranging from 1,000 to 4,000 lb. per hour, the diameter varying from 4 feet in the smallest to 6 feet 6 inches in the largest size, with corresponding lengths of 10 feet and 24 feet. It is probably used principally in the 2,000-4,000 lb. per hour class, since below 5 feet diameter it is somewhat cramped for inspection and cleaning purposes and the internal flue diameter becomes rather small for efficient operation. At 6 feet diameter it is possible to fit two internal flues, as in Fig. 100, and this is the smallest size of Lancashire boiler made.

The maximum size of Lancashire boiler is about 10 feet diameter; the smallest and largest lengths being respectively 18 feet and 30-32 feet, giving an evaporative range of 3,000-12,000 lb. per hour. Cornish boilers are normally constructed for pressures up to 160 lb. per square inch, and Lancashire boilers up to 200 lb. per square inch, although there are examples of Lancashire boilers in service up to 260 lb. per square inch.

In either type of boiler the limiting factor in regard to evaporative capacity is again the grate area, this being curtailed in the smallest size by the permissible diameter of flue fitted inside the shell, and in the larger sizes by the length of grate that can be properly worked by hand.

In a natural draught boiler this restricts the grate area to a maximum of about 50 square feet divided between two flues, and a combustion rate of 25 lb. of coal per square foot of grate area. Higher combustion rates are possible by using mechanical stokers and mechanical draught, but the convective heating surface of a Lancashire boiler is such that the addition of an economiser is necessary to obtain a good overall efficiency.

The horizontal shell class of boiler, however, has the important advantage that a reasonably maintained unit may have a useful working life of upwards of 40 years; its surfaces are readily accessible for inspection and cleaning; and its operation may be entrusted to a low grade of labour, although this is not advised from a fuel economy point of view. Good results on any boiler above the simplest vertical types require a skilled man as fireman.

### HEAT TRANSMISSION IN CORNISH AND LANCASHIRE BOILERS

The ratio of heating surface to grate area is usually from 20 to 30:1, but some 50–60 per cent. of the evaporation of an internal-flue boiler is considered to be produced over the furnace portion, the bulk of this being effected by means of radiant heat. Assuming that the length of internal flue in which the radiant heat effect is most pronounced is approximately 15 feet, then it will be apparent that about 25 per cent. of the total heating surface provided is thus employed.

The remaining 75 per cent. of the heating surface is intended to absorb the heat rejected from the furnace, but this surface must be supplemented by superheater, economiser and/or air heater heating surface if thermal efficiencies of the order of 75 per cent. are desired.

The internal furnace is completely water cooled, allowing large heat-release figures per unit volume, but it also has the disadvantage that any irregularity in the combustion conditions may result in heavy smoke.

The importance of the firebed surface in furthering radiant heat transmission has already been mentioned in Chapter VIII, and since the transmission varies as the fourth power of the absolute temperature, it is worth repeating that a bright, hot firebed must be maintained at all times if the best results are to be obtained.

The use of mechanical stokers, mechanical draught and preheated air is therefore to be encouraged, although the application of these to an existing boiler should be considered in consultation with the makers of such equipment. The heat absorption capacity of an internal flue has limits, and, if these are exceeded, structural defects may develop.

The convective heating surface of a Cornish or Lancashire boiler consists in the main of curved surfaces arranged over external flues, the products of combustion being constrained to flow through the external flues during their passage to the chimney flue. A great deal of heat transmission by gas radiation effects occurs, especially in the first pass below the boiler (where temperatures up to about 1,000° F. occur), and the effects are doubtless assisted by the turbulence resulting from the change in direction of the gases when they leave the internal flues.

Apart from the nature of the surface provided, the design of the external flues does not lend itself to high efficiency. The flues are proportioned to give bodily access for cleaning, etc., as the primary object, and as a result, it is probable that in many boilers layers of gas have only a brief contact—if any—with the external shell on their way to the main flue.

Moreover, the brickwork walls of the external flues must be heated every time steam is raised, and they continue to absorb heat during the steaming period, in proportion to the radiation losses of the setting as a whole.

A defective setting can also cause considerable loss by allowing the gases to be short-circuited directly from the flue below the boiler, into one of the side

flues. A 30-feet long Lancashire boiler expands longitudinally about  $\frac{3}{4}$  inch when heated, and the movement eventually causes spaces to form between the shell and the back end and main seating brickwork, through which the gases flow into the side flues and thence straight into the chimney without giving up heat. Similarly, leakage spaces develop at the front end of the boiler and allow air to be drawn into the main flow of gas, with detrimental results. Cracks in the setting walls have the same effect, the importance of which has already been emphasised in Chapters VI and XI.

### MODIFIED TYPE OF LANCASHIRE BOILER

Defects of this nature have caused at least one British boiler manufacturer to develop an arrangement with a Lancashire type of boiler in which the inefficient external flues are dispensed with. In the modified design the internal flues are slightly smaller and are set rather higher in the shell than is customary, and in the space below them are arranged suitable nests of smoke-tubes, as indicated in Fig. 101. The tubes pass from endplate to endplate, so that the products of combustion discharged from the main internal flues flow through tubes in the water space of the boiler.

The internal flues are proportioned to allow of maximum radiant heat transmission, while the smoke-tube cross-sectional area is such that the gases flow through at a high speed, thus increasing the convective heat transmission.

On leaving the smoke-tubes at the front end of the boiler the gases are divided into two approximately equal streams, and are then caused to flow through tubular air heaters arranged at ground level, on each side of the boiler shell. Most of the remaining usable heat is thus abstracted.

Balanced, forced or induced draught, is employed in this boiler, and an efficiency of 83.3 per cent. is stated to have been obtained in tests. With the exception of the downtake lining (which has a patented air-excluding device) no brickwork setting is required.

### EXTERNALLY-FIRED MULTITUBULAR BOILERS

In externally-fired boilers of the horizontal shell class the internal flues are omitted and the fire-grate is arranged externally below the lower part of the shell, as illustrated in Fig. 102.

The surface normally used for convective heating in the Lancashire boiler

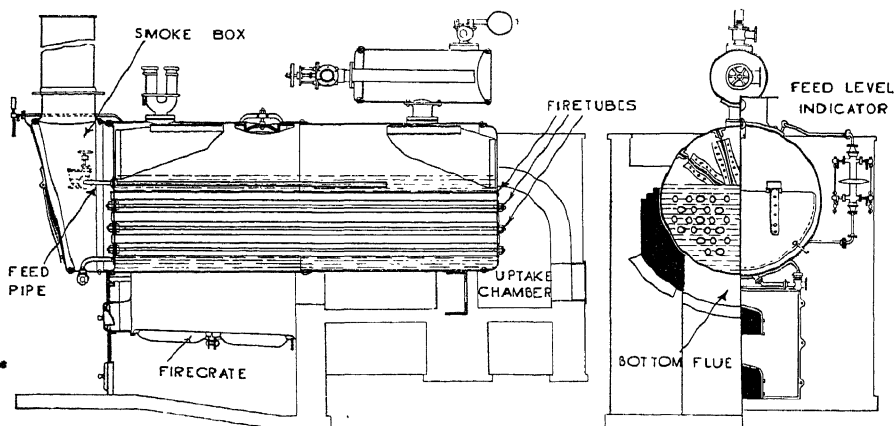


FIG. 102. Externally-fired multitubular boiler.

is in this form used principally for radiant heat transmission. The advantages of this are that the furnace portion can be designed to burn various kinds of waste fuel that could not be readily handled on the restricted area of an internal-flue boiler, and the water circulation along the bottom of the shell is greatly improved.

Usually the externally-fired boiler is mounted upon a brickwork setting, and the products of combustion flow along the bottom of the shell into an uptake chamber, one side of which consists of the back endplate of the boiler.

The back and front endplates are perforated to take smoke-tubes arranged longitudinally along the shell, and the gases flow through the tubes. The discharge from the tubes may be led into a chimney uptake, or guided back along the sides of the shell by means of suitable external flues.

The great disadvantage of the externally-fired boiler is that it requires reasonably good feed water, and, if this condition is not met, overheating of the shell plates along the bottom is almost inevitable when deposits accumulate.

Externally-fired boilers of this class are constructed for evaporative capacities between about 1,000 and 9,000 lb. of steam per hour at 160 lb. per square inch. Ratios of heating surface to grate area in the brick-set type vary from 30 to 70:1; the corresponding ratios of tubular heating surface to shell surface being from 3.5 to 11.5:1.

### THE ECONOMIC BOILER

The externally-fired multitubular boiler has never been employed in Great Britain to the extent that it has in the U.S.A.; British factory engineers apparently prefer to use other forms of horizontal shell boilers.

The disadvantages of the Lancashire boiler in regard to space occupied and its low efficiency necessitating the provision of an economiser led to the introduction of the Economic boiler and many engineers turned to this internally-fired type of boiler as a solution of their problem.

This class of boiler, for a given output, occupies only about one-half the space of a corresponding Lancashire boiler without economiser, and can be constructed as a self-contained unit, without a brickwork setting. A Lancashire boiler capable of evaporating 10,000 lb. per hour, for instance, would be 9 feet in diameter by 30 feet long, whereas an Economic boiler having a similar output would be 9 feet in diameter by only 15 feet 6 inches long.

In making comparisons of space occupied it should be borne in mind that with an Economic boiler space must be provided for cleaning, and, if necessary, replacing the tubes, and this makes the approximate overall length required twice the length of the tube, plus the combustion chamber, and plus again any space required behind the combustion chamber for access purposes. On the other hand the Lancashire boiler requires the overall length of the setting and such flues as come inside the boiler-house, plus some 8 or 9 feet for firing space.

The shorter shell of the Economic boiler also reduces the initial and maintenance costs of brickwork settings, where these are deemed necessary. Very few brick-set Economic boilers are, however, now installed as it has been found that no improvement in either efficiency or evaporation is obtained and maintenance costs are saved. One advantage of brick side flues is that they act as grit arresters.

With these boilers, under good conditions of operation, an average of up to 75 per cent. efficiency can be expected and with experienced stoking and suitable draught to burn the coal, even higher efficiency is obtainable without the use of economisers, air-heaters or superheaters.

The Economic boiler consists essentially of one or more furnace tubes, arranged in a cylindrical shell (see Fig. 103), the gases from the furnaces being discharged into an external combustion chamber. The combustion chamber

is refractory lined to assist the completion of the process. The gases then flow to the front end of the boiler through smoke-tubes passing from endplate to endplate. Usually one, but often two, and sometimes three "passes" of tubes are employed to achieve the maximum convective effect, but it will be appreciated that the multi-pass construction results in greater draught losses, and generally necessitates the application of induced draught.

As in all internally-fired boilers, the evaporative capacity depends on the heating surface and the grate area and the draught conditions to burn the fuel required. The larger sizes of Economic boiler, however, are provided with mechanical stokers working in conjunction with mechanical draught, and up to 40 lb. of coal per square foot of grate area can then be burned, giving an evaporation of about 10 lb. of steam per lb. of coal.

This class of boiler is constructed for evaporative capacities ranging from 750 to 30,000 lb. of steam per hour, and for maximum pressures of 250-300 lb. per square inch according to the size of the boiler. The boilers vary in

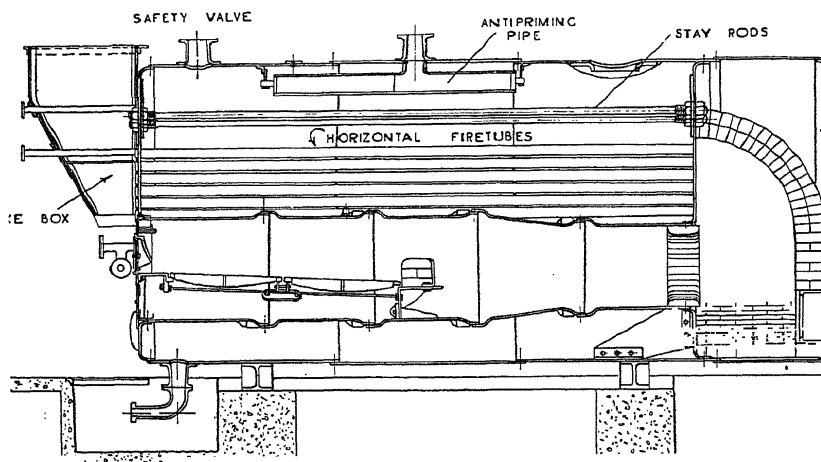


FIG. 103. Economic boiler.

size from 4 feet 6 inches diameter by 7 feet 6 inches long in the smallest, to 13 feet diameter by 20 feet long in the largest, the combustion chamber adding another 2-4 feet to the overall length.

In the conventional design of Economic boiler a single furnace tube is fitted for evaporations up to about 4,000 lb. per hour, and outputs up to 25,000 lb. per hour are obtainable with two furnace tubes. For capacities higher than this it may be advisable to consider the adoption of a water-tube boiler, but if local conditions justify the retention of a shell type, there are Economic boilers capable of generating up to 30,000 lb. of steam per hour with three or four furnaces.

As in other internal-flue boilers, a major portion of the total evaporation is effected over the furnace heating surface, which, in the Economic class, amounts to about 13 per cent. of the total heating surface.

### ECONOMIC BOILER DESIGN

In the later designs of Economic boilers the combustion chamber is made as an integral part of the boiler and is brick lined and lagged with the boiler and thus the radiation losses and brickwork troubles are eliminated. The refractory

lining in the chamber tends to promote more thorough combustion, the process being assisted by the turbulence resulting from the flow of gases around the chamber. Combustion will be completed in the internal flue if the quantity of air admitted to the furnace is correct for the amount of fuel being fed on the grate, and the quantity of fuel supplied is correct for the area of grate. It is desirable that combustion should be completed before the gases reach the chamber. The depth of the chamber should be adequate for the free flow of the gases and their change of direction, this depth depending on the class of fuel used and the size of the boiler for the required evaporation.

The amount of tube area provided for the flow of gases from the combustion chamber is a function of the tube length per diameter ratio adopted, of the ratio between the heating surface and grate area and of the total cross-sectional area required to keep the draught loss within economical limits.

The amount of tube area for the flow of gases from the chamber should be such as to give the correct velocity required for heat transfer. The length of the tube can then be made such that the temperature of the gases is reduced by the desired amount in passing from the combustion chamber to the exit of the boiler.

Generally speaking, a length of 15–16 feet is about the practicable maximum for the ordinary type of smoke-tube. A tube diameter of  $2\frac{1}{2}$ –3 inches is the most suitable to allow for proper tube spacing and to give freedom from rapid choking with soot, bearing in mind that small tubes give a greater heating surface per unit of boiler volume occupied than large ones.

The greatest length per diameter ratio is therefore about 70 : 1. This may be reduced to half this figure in the small sizes of boiler.

The ratio between total heating surface and grate area depends upon the designer's experience and practice, and varies from about 20 : 1 to just over 60 : 1.

The designer's task has been to fit the desired area of heating surface into the smallest (and therefore least expensive) size of shell, while allowing at least 6 inches of clear space around the furnace or furnaces and between the smoke-tubes and shell, as well as about 1 inch to  $1\frac{1}{4}$  inch around each tube periphery. At the same time, he must also provide a suitable ratio of steam space to water space—usually  $\frac{1}{2}$  to  $\frac{3}{8}$ —and ensure that there is sufficient steam disengaging area at the normal working water level in the boiler, or priming difficulties may arise under working conditions.

In addition, the empirically determined ratio of total tube cross-sectional area to grate area must be maintained within the limits of one-tenth to one-fifth common in British practice. This task, of course, has in the past necessitated a considerable number of experiments in laying out specimen tube plate arrangements, etc., and these have now been incorporated in the tables of standard sizes offered by all reputable boiler makers.

## MODIFIED ECONOMIC BOILER TYPES

Within recent years a few manufacturers, however, have modified their standard designs with a view to eliminating a great disadvantage of all internally-fired horizontal shell boilers, i.e. poor circulation along the bottom, in addition to improving the heat transmission generally.

Many of these modifications are not really new, being adaptations of arrangements previously used. They consist principally of placing tubular heating surface in the lower part of the shell, coupled with increased gas velocities on the fire side. Mention can here be made of the two- and three-pass boilers, the latter having a specially designed combustion chamber. The better disposition and use of the available heating surface have undoubtedly resulted in enhanced performance and increased efficiency, which under working conditions is often

of the order of 80-82 per cent. The introduction of a further bank of smoke-tubes necessitates the use of an induced draught fan which, however, enables complete regulation of the draught to be maintained with a lower height of chimney with slightly increased operational costs for the fan power.

One or two makers have produced improved forms of smoke-tube having a sinuous form for which greatly increased rates of heat transmission over a given length are claimed in comparison with the normal plain tube.

One maker has also adopted the completely immersed combustion chamber characteristic of the Scotch marine boiler, thus eliminating the external combustion chamber losses, but he modifies the design by making the boiler of the two-pass type.

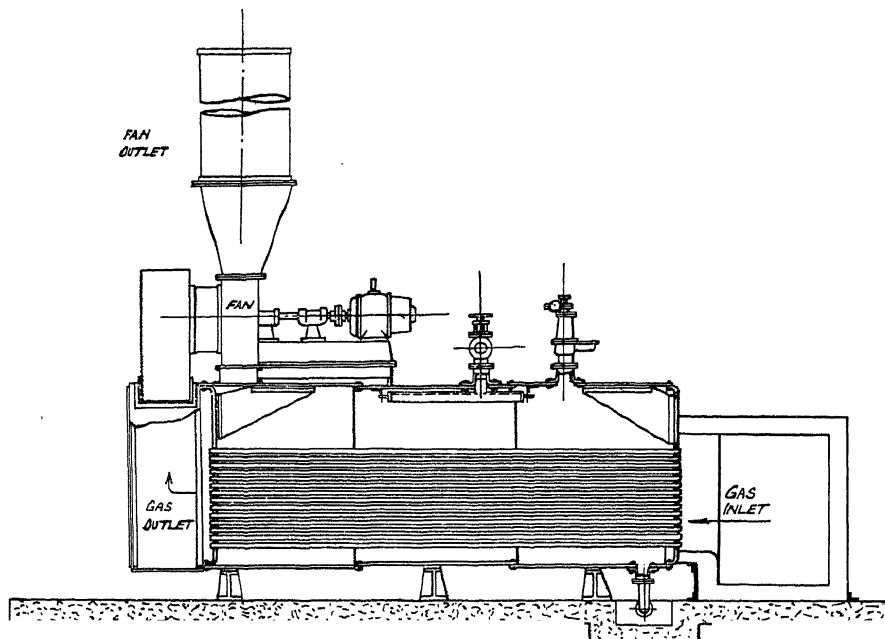


FIG. 104. Waste heat boiler.

## WASTE HEAT BOILERS

Before leaving the subject of shell boilers, mention must be made of the waste heat type. In many manufacturing operations volumes of furnace gases at temperatures of the order of  $1,000^{\circ}\text{C}$ . and less are surplus to the main manufacturing process, and can be utilised for the generation of steam. Boilers of the water-tube class were at first employed for the purpose, but it was found that more satisfactory results could be obtained from a cylindrical horizontal smoke-tube boiler.

In some industries steam raised from waste gases is required at high pressure, that is to say, beyond the limit for which the shell boiler is suitable and the water-tube boiler is still installed to meet these conditions; this also applies where the gas quantities are very large.

The gases to be used are led in special ducts to a refractory-lined inlet chamber and then flow through smoke-tubes expanded into the front and back tube-plates and situated within the water space of the boiler. The cooled

gases are collected in an outlet chamber before being drawn off to the chimney. A diagrammatic view of a waste heat boiler provided with an induced draught fan is shown in Fig. 104.

Since the temperature and the hourly volume of the gases available are capable of almost infinite variation over the field of industry, the standardisation of such boilers is somewhat difficult. Each boiler must therefore be designed with particular conditions in mind, but there are a few fundamental factors that are common to all tubular waste heat boilers.

The heat transmission is accomplished chiefly by convection, so that the insulating effect of gas films on the tube walls must be overcome by increased gas velocity. A fairly high gas pressure drop across the boiler is necessary to achieve this object, and an induced draught fan is thus an almost essential auxiliary. The sudden increase in velocity as the gas enters a tube mouth results in considerable additional turbulence extending for some distance into the tube, the extent of the effect depending upon the magnitude of gas pressure drop considered to be permissible in any given circumstances.

The ratio of tube length to diameter is also of importance as the gas must be allowed sufficient time to impart its heat to the water, and ratios up to 150 : 1 have been applied. In modern boilers the ratio may be in the neighbourhood of 120 : 1, but when sinuous tubes are fitted it may be reduced to as low as 66 : 1, with consequent savings in boiler size and constructional cost.

Again, since evaporation takes place on the wetted surfaces of the tubes, space must be allowed for the free release of steam bubbles and the unretarded circulation of water. The tubes are in consequence fairly widely pitched, horizontally and vertically, and are provided with circulation lanes extending the full depth of the shell. Ample area is also required between the peripheries of the outer tubes and the shell.

Within the last decade or so boilers of the thimble-tube and forced-circulation coil types have been used for waste heat utilisation, but as these applications are in the main in connection with internal-combustion engines, they will not be discussed here. Waste heat boilers are discussed in detail in Chapter XX.

## SHELL AND WATER-TUBE BOILERS

The shell boiler, because of its constructional characteristics, is normally limited to a maximum working pressure of about 250 lb. per square inch, although a few have been constructed for 300 lb. per square inch; any higher pressure imposes the use of abnormally thick shell plates, when maximum evaporative capacity is also required. Similarly, shell boilers are generally restricted in land practice to an output of about 20,000 lb. per hour in a single boiler, although in marine practice the Scotch type of multitubular boiler is designed for outputs in excess of that figure.

The water-tube boiler can be constructed for capacities varying from a few hundred to more than a million lb. of steam per hour, and for all steam pressures up to the critical value of 3,200 lb. per square inch.

For the smaller steam generating plants in making the choice between shell and water-tube boilers for an installation, consideration would need to be given to efficiency, relative first and operational costs, working pressure required, load fluctuations and local conditions. For large installations, however, where high pressures and large outputs are required, the selection must inevitably fall on the water-tube boilers.

Before leaving shell boilers, mention should be made of the considerable advance which has been made in recent years in the design and operation of furnaces and stokers for horizontal shell boilers. These have undoubtedly contributed to the high efficiencies reached, enabling coals of a very wide



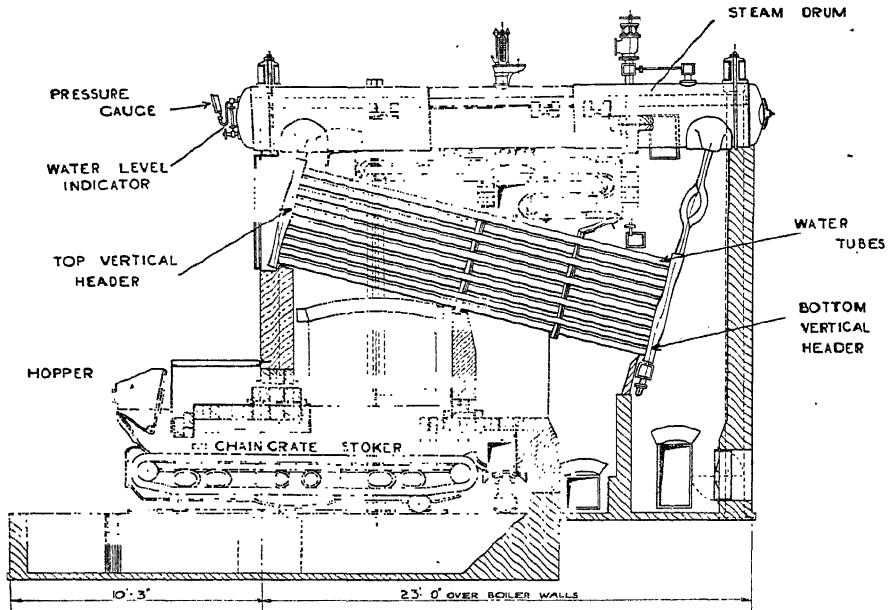


FIG. 105. Water-tube boiler—straight tubes with sectional headers.

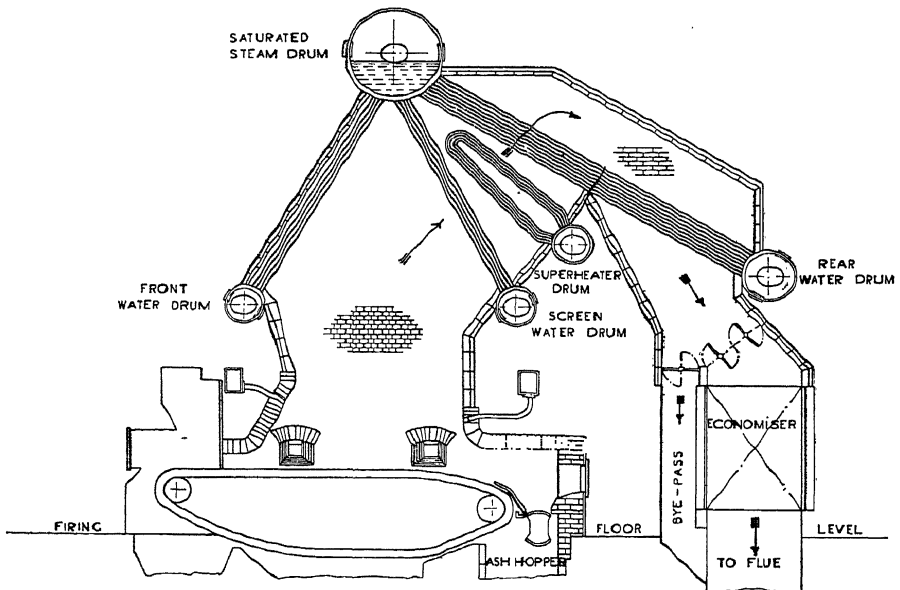


FIG. 106. Water-tube boiler—tubes expanded into drums.

## STEAM BOILERS

range, both as to quality and grade, to be used in them. Mechanical stokers have also greatly simplified the problem of firing large boilers.

Shell boilers are very easily adapted for oil firing, and very many of them are installed for heating institutions, large buildings and workshops, the boilers being used for generating hot water at high or low pressure or steam ; some are fired by gas or electricity.

### WATER-TUBE BOILER TYPES

There are three principal types of water-tube boiler used in this country at the present time ; the straight-tube, the bent-tube and what might be termed the forced circulation boiler.

The first is exemplified in Figs. 105 and 106 ; the former being a sectional

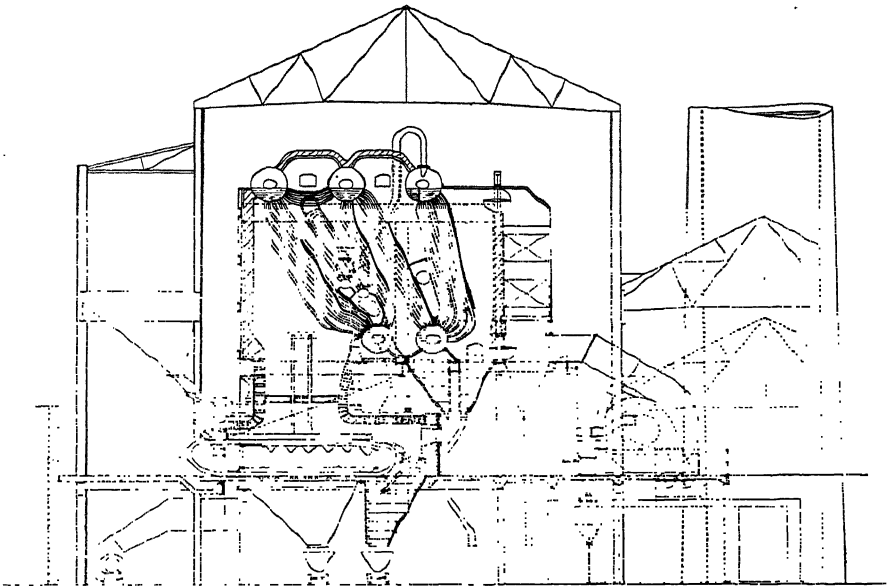


FIG. 107. Water-tube boiler with five drums.

header straight-tube boiler and the latter a vertical straight-tube boiler with tubes expanded into suitable drums.

The second type, the vertical bent-tube boiler, is shown in the five- and three-drum arrangements in Figs. 107 and 108 ; the number of drums may be reduced to two or increased to four or more, according to the purpose for which the boiler is intended.

The third type is to be found in boilers such as that shown in Fig. 109, the heating surface of which consists principally of water tubes suitably connected to form a continuous length.

### EFFECT OF PRIME MOVER ON BOILER DESIGN

The development of the water-tube boiler from the comparatively small type of unit in use at the beginning of the present century to the modern high-pressure, high-capacity steam generator, has resulted largely from the demands made by steam-turbine designers for an increased heat content per lb. of steam flowing through the turbine stop valve.

The thermal efficiency of a steam turbine depends upon the difference between the initial and the final temperature of the steam supplied to and exhausted from the machine, and since the lower temperature is at its economic

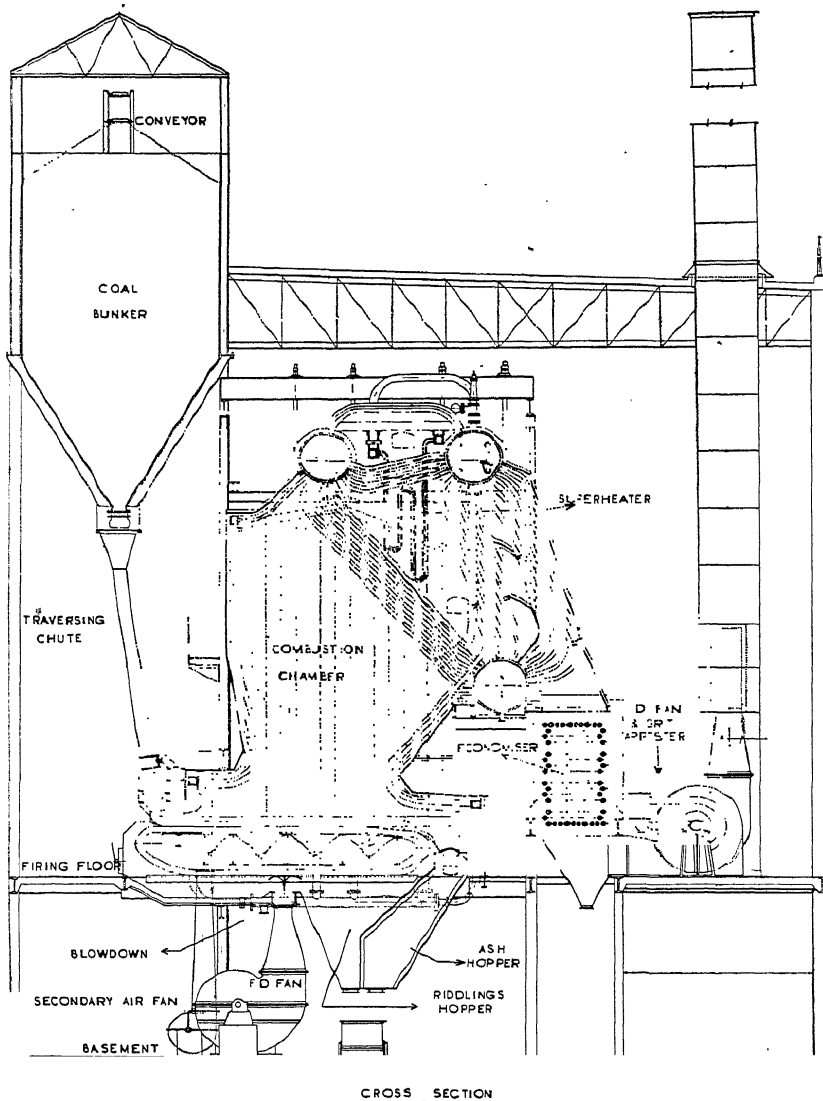


FIG. 108. Water-tube boiler—bent tubes with three drums.

limit, improvement has had to be obtained on the boiler side. An increase in steam pressure raises the initial temperature, but the usefulness of higher pressures is limited by the amount of condensation permissible at the turbine exhaust, the moisture content in good practice being about  $12\frac{1}{2}$  per cent.

The stage in a given expansion range at which condensation begins can be retarded by the imposition of superheat, or by re-heating the steam, so that pressures up to the critical value and steam temperatures up to 1,000° F. have been applied in service. It will be appreciated, however, that unless the steam is re-heated between the high-pressure and low-pressure stages of the turbine, a definite relationship of initial steam pressure to initial total steam temperature

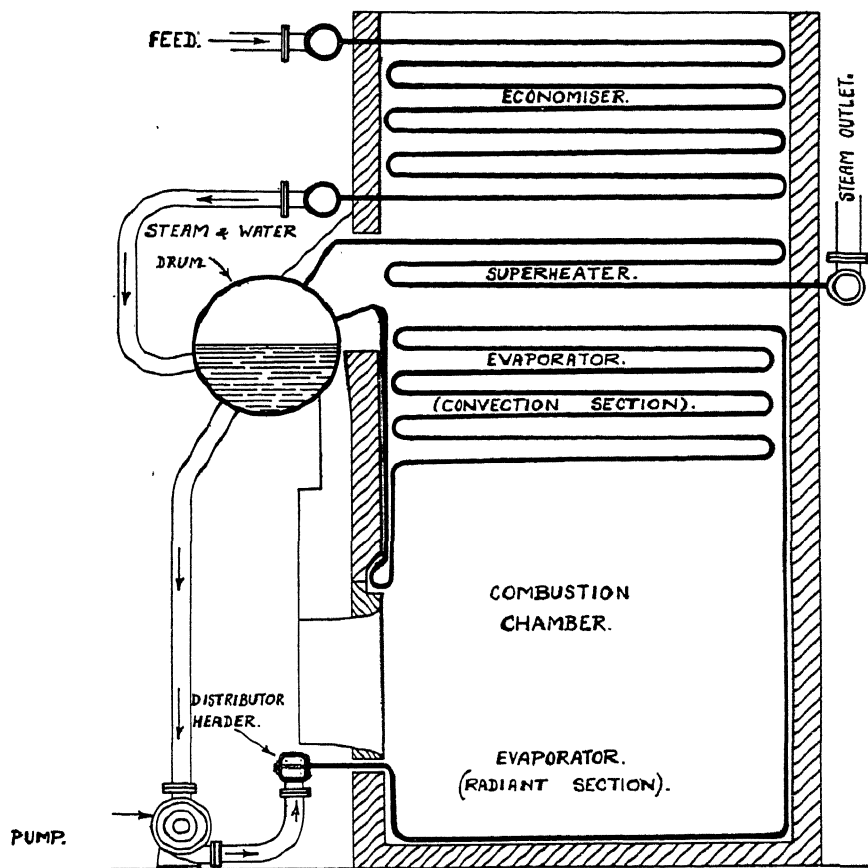


FIG. 109. Water-tube boiler—forced circulation.

must exist if excessive condensation at the turbine exhaust is to be avoided. Sir Leonard Pearce recently stated that for normal turbine conditions the steam temperature and pressure relationship is approximately as set out below.

Steam temperature ° F.			Steam pressure lb. per sq. in.	
750	..	..	..	580
800	..	..	..	750
850	..	..	..	950
900	..	..	..	1,200
950	..	..	..	1,500
1,000	..	..	..	1,900

To meet turbine requirements, steam pressures and temperatures have been gradually increased until at the present time boilers working at 650 lb. per square inch and 800° F., are relatively common in British power-station and industrial practice. Higher pressures and temperatures than those cited are also in use, but the properties of the commercially available superheater tube materials render it advisable to halt at the 825°–850° F. steam-temperature limit for the time being. Even so, the cumulative effect of these increases, together with the necessity for steam generators of greater capacity to supply the larger steam turbines being built, has been to modify considerably most of the previously accepted designs.

The greatest rate of heat transference can be obtained by the use of radiant heat, and water-tube boiler designers have therefore tended to arrange for the major part of the heat available to be absorbed by a comparatively small amount of "high-grade" heat-receiving surface massed about a high-temperature furnace zone. The area of "true" boiler convective heating surface required has in consequence diminished, and its main function has been delegated to the economiser heating surface, as explained below. The increase in the final steam temperature has necessitated the provision of larger superheaters arranged nearer to the furnace zone to obtain the requisite temperature gradient, and in some examples the superheaters are of the radiant-heat type, lining the furnace walls.

### FEED WATER TEMPERATURE

Another problem that has been transferred to the boiler designer during the process of improving turbine efficiency, is that of high feed water temperatures. By extracting or "bleeding" steam from a turbine at one or more points during expansion of the steam, and using the bled steam to heat the boiler feed water, the thermal efficiency of the machine is raised, and usually the size of the low-pressure turbine can be reduced. In practice, the number of extraction heaters varies from one in a small steam turbine to five in a large one.

The result, in any arrangement, is that the feed water requires less heat from the flue gases to raise its temperature to that corresponding to ebullition. But, for the optimum boiler efficiency of large units installed in central power stations, it is necessary to discharge the flue gases at a temperature of about 300° F. so that a difficult situation arises when the turbine designer returns the condensate to the boiler house at a temperature of anything up to 400° F.

In modern designs this difficulty has been overcome by a reduction in the amount of true boiler convective surface provided; thus, in effect, moving the economiser nearer to the furnace. The economiser heating surface is then swept by gases at a temperature sufficiently high to provide the gradient necessary for heat transference to the water.

The gases leaving the economiser have still a large amount of usable heat, and this may be absorbed by an air heater installed in series with the economiser. The use of an air heater in these conditions, however, is limited by the method of burning the fuel. Mechanical stokers do not operate satisfactorily when the air temperature exceeds about 350° F., and the upper limit for primary air for pulverised fuel or oil burning is in the region of 500° F.

Thus the adoption of regenerative feed water heating may at times force the boiler designer not only to re-arrange the disposition and purpose of the boiler heating surface, but also to adopt another method of burning the fuel.

### FURNACE DESIGN

The increase in the evaporative capacity of individual boilers occasioned by the use of larger turbo-generators has reacted on the dimensions of water-tube boilers, particularly in the furnace. The earlier boilers were constructed

with their heating surface arranged at only a short distance from the fire, and any attempt to raise the heat input was limited by smoke and similar troubles. In Chapter VIII it was shown that the rate of radiant heat transmission in a furnace is directly related to the fourth power of the absolute flame temperature, and that the percentage of the available radiant heat that is absorbed by the boiler heating surface decreases with the weight of air supplied to the combustion chamber per lb. of fuel.

Thus the aim of the designer is to provide as high a flame temperature as possible by restricting to as near the theoretical minimum as possible the amount of air supplied for combustion. It was found that higher flame temperatures and better combustion resulted from an increase in the height of the furnace, but as the heating surface was progressively withdrawn and furnace temperatures rose, trouble was experienced due to the relatively rapid failure of the refractory lining at the higher temperatures.

This aspect of the problem was partly obscured for a time by the construction of larger furnaces, by the use of air-cooled walls and by modifications of the furnace flame, but it re-appeared as furnaces were again forced to the limit of their heat-release capacity.

### ASH AND SLAGGING DIFFICULTIES

Higher furnace temperatures also introduced ash and slagging difficulties, as in many plants the ash in the coal was heated to a temperature above its fusion point. In consequence, the molten ash was deposited on the furnace walls, and either fluxed with the refractory or "froze" on it setting up a form of rapid surface disintegration. Thus, although a furnace lining might be capable of standing up to a furnace temperature of 3,000° F. (1,650° C.), the use of coal having an ash fusion point of, say, 2,000° F. (1,100° C.) would limit the heat input to the rate corresponding to the lower temperature in order that heavy brickwork maintenance costs might be avoided. Similar troubles were experienced in the first rows of boiler tubes due to slag deposits causing choking of the gas passages and a reduction in the rate of heat transmission.

The furnace temperature can be diminished by increasing the amount of excess combustion air, or by providing a larger area of heat-absorbing surface in the furnace. The first method can be applied to practically all boilers, but it gives a lower CO<sub>2</sub> percentage in the products of combustion and a reduction in combustion efficiency.

The second method must be used with discretion, as with low-volatile and similar fuels undue cooling of the furnace may render it difficult to maintain ignition. Water-cooled furnace walls of the bare-tube or block-covered types have therefore been widely adopted in suitable conditions. Existing water-tube boilers will in consequence show improved performance when provided with water walls, but the effect of these on furnace efficiency and the utilisation of the available heat input must not be overlooked.

At the present stage of water-tube boiler development the position is that the necessary conditions of turbulence, temperature and time must be provided in the furnace to obtain the best all-round results, but the methods of meeting these conditions vary according to the class of coal burned and the type of boiler employed.

### WATER CIRCULATION

The addition of water walls and the increases in steam pressure and feed water temperature described above have introduced new factors in regard to water circulation throughout the boiler. A free natural circulation of water presupposes a reasonable difference in temperature between one part of the

circulatory system and another and boiler engineers have had to take care of such physical characteristics when designing the various sections of a boiler.

Special consideration needs to be given to the separation of the steam from the water in regions subjected to high rates of heat transmission. Designers have therefore given considerable attention to the provision of adequate downcomer (or feed) area and the correct distribution of water to the various heating surfaces. These design problems have been solved to such an extent that natural circulation boilers in commercial undertakings have been operating satisfactorily for extended periods at pressures up to 2,500 lb. per square inch.

With forced circulation, as in the type of boiler illustrated diagrammatically in Fig. 109, there is a controlled circulation of water under all conditions of operation, and the heating surface can be arranged to obtain the maximum heat absorption. Moreover, the diameter of the water tubes can be made smaller and the design itself can be conveniently adapted to restricted space conditions at site. There may also be some advantage in reduced weight of the boiler.

A number of these boilers are in service in this country, but some engineers consider that they introduce complications in the form of continuously-running circulation pumps and difficulties in the removal of scale from the interior of the water-tubes.

Although much of the foregoing discussion applies to water-tube boilers at high evaporative capacity, the experience gained in extensive development work has been used in the design of the relatively smaller boilers installed in numerous factories.

Advantage has been taken of the increased knowledge of heat transmission to produce water-tube boilers capable of working efficiently within the Economic boiler range of evaporation. Some very compact steam generators have thus been made available for industrial service. In all installations the feed water supply must be reasonably pure.

Again, existing water-tube boilers have been modified to bring them into conformity with modern ideas on furnace design, and stoker and pulverised fuel burner manufacturers have collaborated with boiler designers to produce efficient fuel-burning appliances suitable for the various heat inputs required.

Progress has been facilitated by the greater use of boiler-house instruments capable of continuously checking and recording combustion efficiency and which give immediate warning of any variation from normal performance.

However, in order to secure the full advantages from the installation of more efficient boiler units and also from provision of a full range of boiler house instruments, it is essential that skilled operating engineers be placed in charge of the plant.

## CHAPTER XIV

## STEAM BOILER AUXILIARY PLANT

Boiler mountings and fittings—Soot removal—Superheaters—Economisers—Feed water heaters—Air preheaters—Auxiliary power and steam utilisation—Boiler feed pumps—Pumping hot water—Maintenance of feed pumps—Draught production by fans.

**T**HE performance of a steam boiler, as has been indicated in Chapter XIII, depends upon the correct disposition of its heating surface, and for the highest efficiencies it is necessary to superimpose on a sound design additional heat-absorbing appliances, such as superheaters, economisers and air preheaters. Certain mountings, fittings and instruments must be attached to the boiler installation. Similarly, the selection of feed pumps, fans and other auxiliaries has an important effect on the efficiency of the boiler plant as a unit.

## BOILER MOUNTINGS AND FITTINGS

Attachments to boilers are divided fairly rigidly into two classes : mountings and fittings. Instruments are a separate item and seldom actually attached to the boiler fabric.

Under the heading of mountings are comprised safety valves, high and low water alarms, main stop valves, feed and check valves, circulators, blow-down valves, water gauge frames and valves, pressure gauges, anti-priming pipes and scum valves.

Under fittings would be included damper frames and damper gear, injectors, soot blowers, furnaces and stokers, superheaters or preheaters, feed regulators and thermostatic or pressurestat controllers.

Generally speaking the term "mounting" implies that the equipment is mounted on a pad or stool riveted on to the fabric of the boiler, as distinct from a "fitting" which may or may not be attached to the boiler, but for which there is neither stool nor pad.

In both categories there are some mountings and fittings which are essential to operation and others which may not be essential, but which in themselves, in one form or another, fulfil a perhaps necessary service to the plant to which they are attached.

*Safety Valves.* Safety valves are essential on all steam boilers, and are generally of three main types : dead weight, lever or steel yard and spring loaded, each type being made in a variety of patterns with or without means for locking or adjustment. Whichever type is selected the design should be such that unauthorised interference with the setting of the valve is impossible.

Every boiler must be fitted with two independent safety valves each capable alone of discharging the maximum amount of steam that the boiler can produce ; so that if one valve sticks for any reason there is still one left functioning and the danger from abnormal pressure is avoided.

Unless it is imperative to do so, discharge through a safety valve should never be piped away ; it is far better that the safety valve should only blow on very rare occasions. Every time the valve blows, fuel is wasted at a prodigious rate.

*High and Low Water Alarms.* A very common valve of this type comprises a balanced float gear which is usually built into the lever type of safety valve. The combination thus provides a safety valve as demanded by law, together with high and low water protection. The sound made by the escape of steam is different from that made by water, so that the operator can detect whether it is the high or low water alarm which is functioning or whether it is the safety valve which is blowing off.



*Boiler Feed Valves.* The boiler feed or "clack" valve probably does more work in continuous movement than any other piece of equipment on the boiler, particularly if the feed pump is of the reciprocating type. The correct position for the feed valve pad is about 3 inches below the water level, and it should be set as close to the boiler shell as possible so that there will be no necessity for a bend in the internal feed pipe to carry the feed into the internal shell in a downward direction. Internal feed pipes in shell boilers should not be extended beyond the middle of Lancashire or Cornish boilers, and in the Economic boiler the pipes should never reach within 6 feet of the rear tube, especially if the feed is relatively cool.

*Circulators.* There are two major types of circulator, (1) thermodynamic, depending on convection currents, and (2) the static which makes use of energy otherwise dissipated in priming. All shell boilers suffer from faulty circulation owing to the fact that heat is always applied at a point above the lowest water level, so that circulators are added to overcome this difficulty in some measure.

Static circulators are characterised by a hood or reaction plate which is always placed in the position where the maximum priming occurs. Circulators require that the water level should be rigidly maintained, otherwise, if over-feeding takes place, violent priming in the boiler might result.

*Blow-down Valves.* Blow-down valves are an important fitting to any boiler and should be essentially of first class structure, inasmuch as the valve is often in a position where it is not continuously under the eye of the operator; leakage through this valve may result in a heavy waste of fuel.

## SOOT REMOVAL EQUIPMENT

A factor having considerable influence on the efficiency of a steam boiler during operation is the thickness of deposits which form on the fire side of its heating surfaces. When boilers are worked at comparatively low rates the deposits may amount to no more than a film of relatively easily removed soot, but in larger water-tube boilers in which furnace temperatures are high, the deposits may assume the properties of a hard cement-like scale. This operating problem has given great trouble within recent years and is not yet fully solved. These notes are concerned more with the method of removing the deposits than with the mechanism of their formation.

Obviously, one way of removing these deposits is to shut down the boiler and scrape and brush the heating surfaces. This is not always convenient, particularly during emergency conditions, and resort is therefore made to the various types of blowing jets that have been developed for this purpose over the past twenty-five years.

In the vertical boiler with water-tubes across the firebox, a nozzle can be arranged below or between the tube nests to play on the spaces between the tubes and blow the accumulated soot up the chimney.

In Lancashire and similar boilers, steam jets can be arranged in the external flues, the soot and dust thus removed being deposited in a special dust-collecting chamber. Such jets work in a comparatively low-temperature atmosphere, but in Economic boilers, where the blowers project into the combustion chamber, to clean the smoke-tubes blowers of the withdrawable or retractable type are necessary if a reasonable working life is to be obtained from them. The blowers are then designed so that a single handwheel causes the jets to move into the combustion chamber and deliver steam at the same time, while oscillating over a pre-determined arc to cover all the tubes.

Blowers of this type can be adapted for use in connection with the furnace and superheater tubes of water-tube boilers, and for other positions where the nozzle can be arranged some 2 feet 6 inches to 5 feet from the area of impact. Where situated in regions of high gas temperature, the blower parts must be

made of suitable materials and must be designed to retract well out of the influence of the hot gases when not in use. A fairly high steam pressure, 200–500 lb. per square inch—is required for the successful operation of this blower.

In regions of lower temperature and when nests of tubes have to be cleaned, a form of multi-nozzle blower may be installed. The nozzles or jets are arranged to blow into the spaces between the tubes, but care must be taken to prevent impingement of the steam-blast on to the thin water-tube walls. These multi-nozzle blowers may be fixed a few inches or up to 2 or 3 feet from the tubes to be cleaned, and the steam pressure required varies from about 100–350 lb. per square inch, according to the distance from the tube nest.

Automatic methods of soot blower operation are now available in which the sequence and duration of action of soot blowers arranged throughout the boiler structure can be determined by push-button remote control, and in large modern water-tube units the soot-blowing equipment is considered simultaneously with the general design.

## HEAT LANCING

In circumstances in which deposits build up on the fire side of water-tube boiler surfaces quicker than they can be removed by the normal design of soot blower, it may be necessary to use a portable steam “gun,” operated by hand.

Recently water sprays have been used working under a pressure of 50 to 100 lb. per square inch. In this system the water is sprayed on to the boiler tubes, and it is stated that after only a few minutes the deposit is loosened. The water is then shut off for an interval until the deposits heat up, and a second spraying usually causes them to break away. The action is somewhat similar to the heating and rapid cooling of copper evaporator coils adopted by the marine engineer to remove scale from the coils without having to open up the evaporator.

Compressed air has not been used a great deal in this country for soot blowing purposes, but it is significant that a recent (1942) American high-pressure power station has adopted compressed-air soot blowing at a pressure of 250 lb. per square inch. The designers of this station consider that the compressed-air system is just as economical as steam blowers, and is more adaptable for combining with a water system, should water jets become necessary.

## SUPERHEATERS

The use of superheated steam has been considered in detail in Chapter VII, where its value for power generation was indicated.

Most superheaters are installed within the boiler setting and receive their heat from the same flue gases and same furnace as the boiler. These are termed “integral” superheaters, and they are generally of the convection type, absorbing heat from the gases of combustion which sweep over them. In some instances externally or separately fired superheaters have to be provided, but in general steam-raising practice these are comparatively rare and for the present purpose can be disregarded.

*Gas Temperatures and Velocities.* For effective heat transmission in a superheater the temperature difference between the gas and steam sides of the elements must be about 300° F. (149° C.). In the convection type of superheater the resistances to heat flow are the gas film, the metal wall of the tube and the steam film. The most important of these is the gas film, the effect of the other two becoming relatively insignificant as the rate increases. The final steam temperature is a function of the gas temperature and of the mass velocity of the gases (i.e. the weight of gases in lb. per second per square foot of gas passage area), a small increase in the mass velocity having a propor-

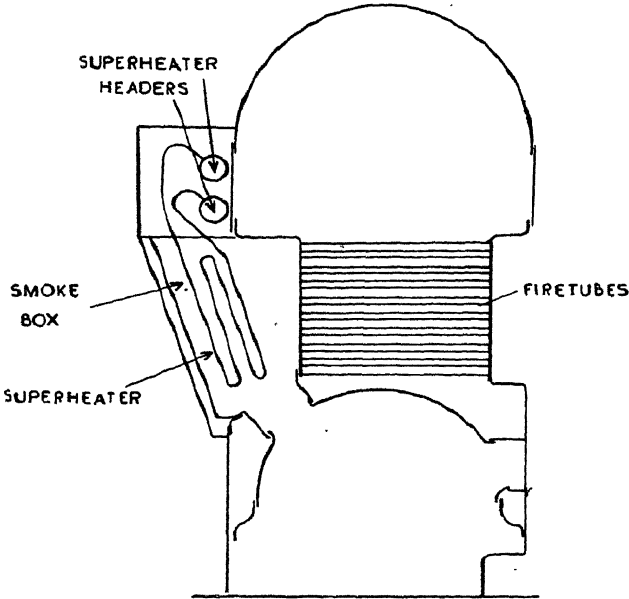


FIG. 110. Superheater applied to a vertical boiler.

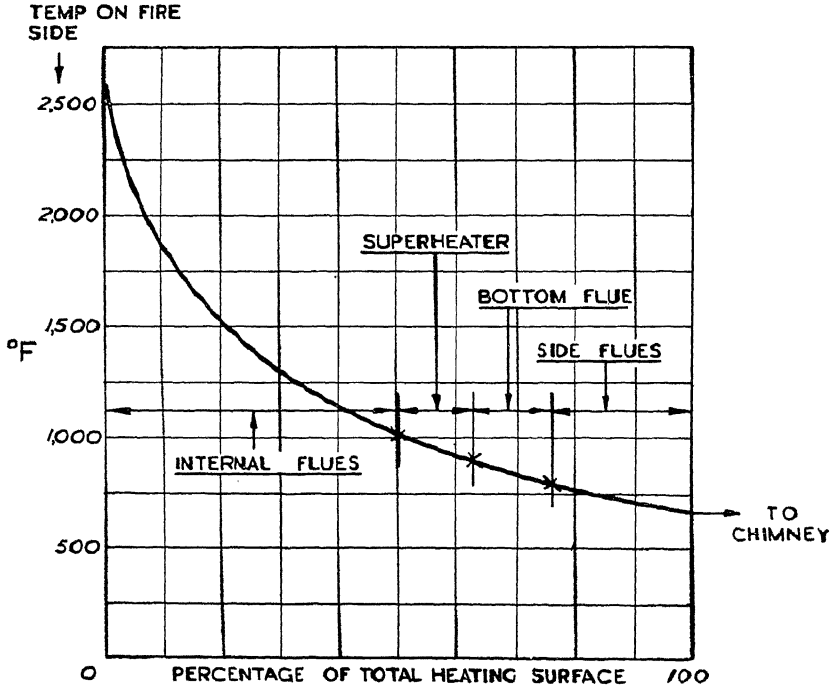


FIG. 111. Temperature curve for a Lancashire boiler.

tionately greater effect on heat transmission than a small increase in gas temperature.

On the steam side, the primary requirement is that all the superheater tubes should receive a uniform flow of steam, and the steam velocity must be high enough to prevent overheating. There is a critical velocity at which heat transmission is at a maximum, according to the tube diameter and length, but too high a velocity may result in an unduly large pressure drop across the superheater. A pressure drop of from 2 to 5 per cent. is usually considered to be permissible, although superheater designers frequently assert that this is unnecessarily low in view of subsequent steam pipe-line losses.

*Superheater Types.* The construction, size and position of a superheater relative to the boiler surface which it amplifies depend upon the steam tem-

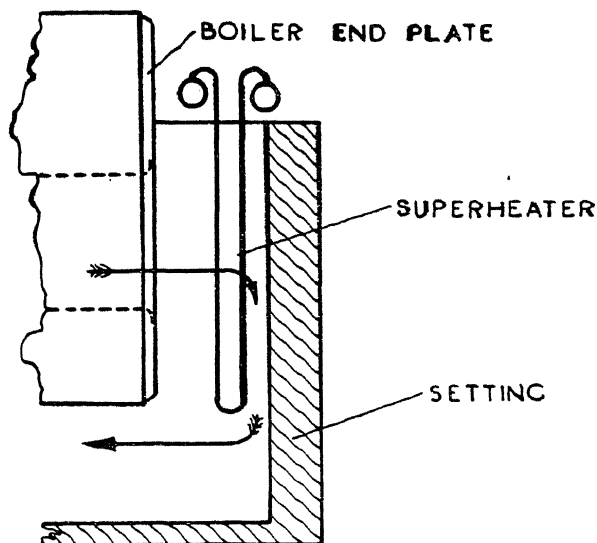


FIG. 112. Superheater arranged for a Lancashire boiler.

perature desired, and also, to a certain extent, upon the type of boiler to which the superheater is applied. In a vertical cross-tube boiler, for instance, the space available for a superheater is extremely limited, and the only practical position is in the uptake. The diameter of the uptake restricts the area of superheater surface that it is possible to provide, so that when fitted in this type of boiler it forms, in effect, an extension of the steam pipe placed in the path of the products of combustion. In a vertical boiler having vertical smoke-tubes, the superheater can be arranged over the top tube-plate in the form of a continuous coil, and the provision of a greater area of superheater surface is possible.

Provision for still greater area can be made when a boiler of the type shown in Fig. 110 is under consideration, and in this example the superheater may be constructed of a number of "hairpin" bends located in the combustion chamber space. This design results in much higher superheat temperatures than those mentioned above.

In a Lancashire boiler the temperature on the fire side of the boiler falls progressively from the furnace to the main chimney flue opening. An approximate temperature curve for a 9 feet 6 inches diameter by 30 feet long Lancashire

boiler is shown in Fig. 111, and it will be seen that the greatest temperature drop occurs in the internal flues.

When the maximum amount of superheat possible with a Lancashire boiler is required, it is necessary to arrange the superheater elements inside the internal flues, where they are subject to gas temperatures of the order of 1,300° F. (704° C.), as well as to a certain amount of radiant heat.

For more normal requirements up to about 250° F. (120° C.) of superheat, the superheater can be arranged at the mouths of the internal flues, as shown in Fig. 112, the gas temperature in this region being about 1,000° F. (538° C.). From the curve it will be appreciated that the remainder of the available temperature drop is of comparatively little use for the higher range of superheat temperatures, since on passing through the bottom external flue the temperature falls to about 750° F. (400° C.).

*Economic Boiler Superheaters.* In an Economic boiler the furnace temperature may be about 2,600° F. (1,427° C.). This falls rapidly along the furnace length until it is approximately 1,600° F. (870° C.) at the furnace exit mouth, a further reduction to 1,300° F. (704° C.) occurring in the combustion chamber. At the entrance to the smoke-tubes a temperature of 1,000° F. (538° C.) is common, and there is then a gradual diminution to possibly 575° or 550° F. (302° or 288° C.) at the uptake end of the tubes, if the boiler is of the self-contained type.

When only a few degrees of superheat are required as, for instance, where steam has to be transmitted over a distance for process purposes, a superheater in the smoke-box will serve the purpose.

For higher superheat temperatures, the superheater can be constructed in the form of a series of U-tubes inserted into the smoke-tubes from the smoke-box end, the saturated and superheated steam headers being arranged in the smoke-box space. A further increase in steam temperature can be obtained by extending the ends of the U-tubes into the combustion chamber, but for the highest steam temperatures possible in the Economic boiler the U-tubes are replaced by a number of single tubes which connect with a series of superheater elements arranged in the upper part of the combustion chamber. Alternatively, the element "bundles" can be connected to headers arranged outside the back-end and casing or setting of the boiler.

The U-tube form of smoke-tube superheater can be applied also to the stationary locomotive boiler, but in this application some of the upper rows of smoke-tubes are replaced by tubes of larger bore capable of receiving the elements, the header or headers being placed in the smoke-box. When only a moderate increase in steam temperature is desired, this space can be utilised to house superheater surface consisting of continuous tubing of suitable size.

*Water-tube Boiler Superheaters.* In water-tube boilers, steam temperatures up to 1,000° F. (538° C.) have been maintained in service, but in normal installations the steam temperature is restricted to about 850° F. (454° C.) to allow of the maximum use of ordinary commercial steel tubes. The degree of superheat obtainable depends upon the position of the superheater in the gas flow and upon the area of heat-absorbing surface interposed between the superheater and furnace.

There are four principal types of superheater employed with water-tube boilers: the interpass, the interdeck, the intertube and the radiant heat superheater. The interpass design is illustrated in Fig. 113, and receives most of its heat by convection. It is generally used in connection with a refractory-lined furnace when a moderate steam temperature is required, and obviously the superheat obtainable can be varied in the design stages by suitably moving the first-pass baffle to expose more or less water-tube area to the flow of gases.

A defect of the convection superheater just illustrated is that the steam

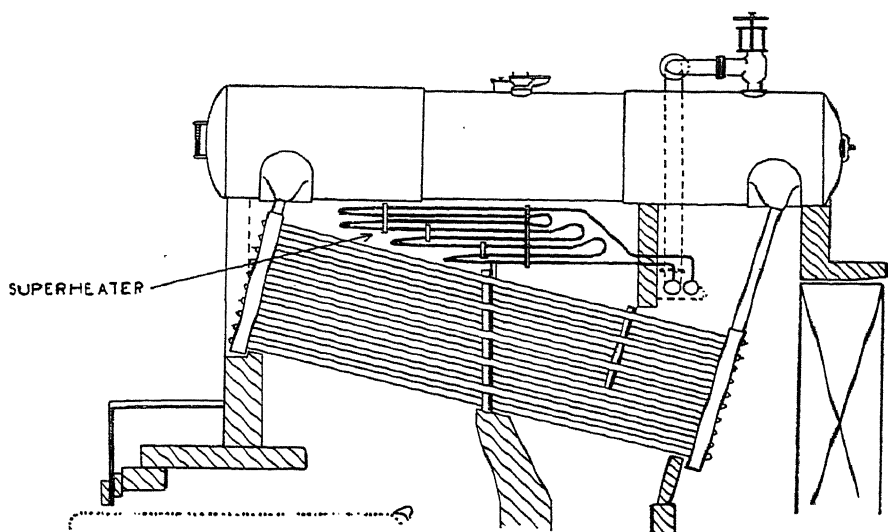


FIG. 113. Interpass superheater in a water-tube boiler

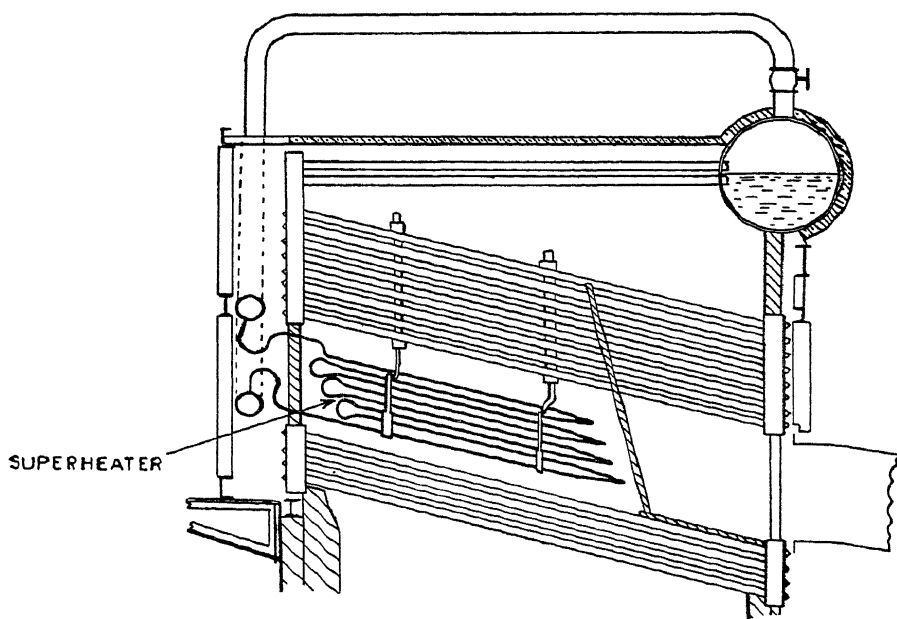


FIG. 114. Interdeck superheater.

temperature rises with an increase in boiler load, so that when a fairly steady steam temperature is essential, an interdeck superheater may be used instead.

As will be seen from Fig. 114; only a few rows of boiler tubes are interposed between the interdeck superheater and the furnace, but these are sufficient to protect the elements from excessive radiation, while allowing a certain amount of radiant heat absorption. Further protection can be afforded by introducing saturated steam into those elements of the superheater that are nearest to the furnace. Thus the superheater works under conditions similar to those of the

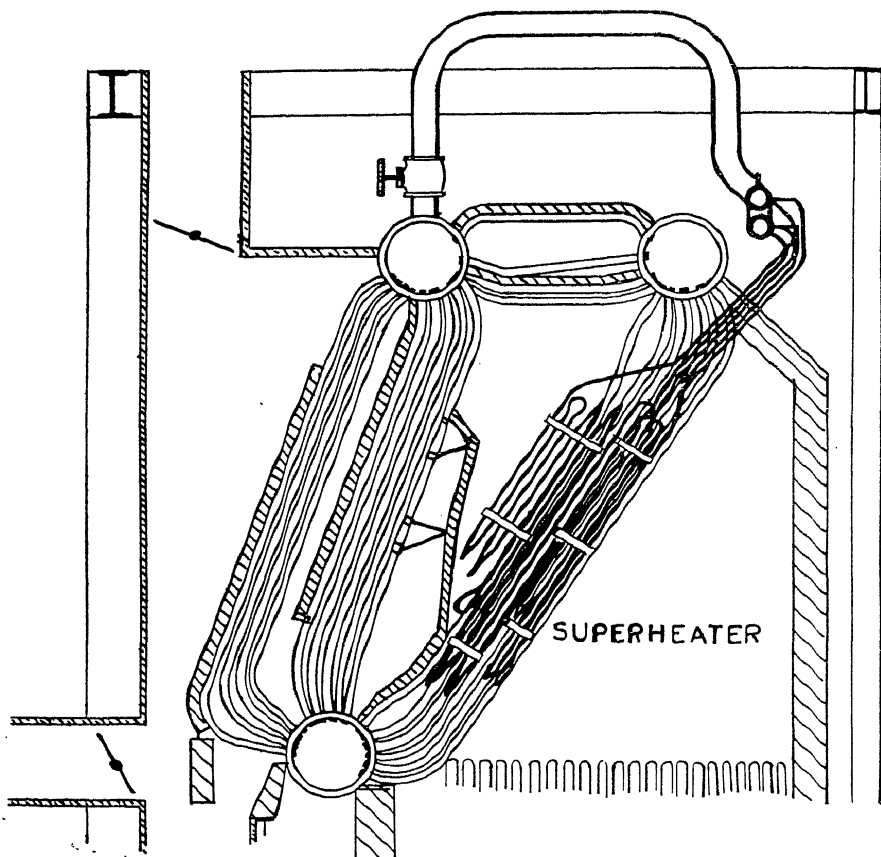


FIG. 115. Intertube superheater.

bottom rows of steam-generating tubes, and variations in the boiler load do not lead to as great a variation in steam temperature as when a purely convective superheater is used.

The interdeck is very suitable for boilers having water-wall furnaces which absorb a large proportion of radiant heat, and the intertube design illustrated in Fig. 115 is used in connection with boilers of the Stirling and related types working under similar conditions. It will be appreciated that the less water-tube surface between superheaters of the interdeck or intertube types and the furnace, the less will be the area of superheater surface required, and the higher

will be the final steam temperature. The more severe the fire side conditions, however, the better must be the quality of the material used for the elements.

**Radiant heat Superheaters.** These remarks are particularly applicable when the water-tube screen is removed and the superheater becomes a radiant heat type. In applications of this nature the superheater tubes may be arranged in the furnace walls.

A characteristic of a radiant heat superheater is that the superheat temperature falls as the boiler load increases, the steam flow increasing at a greater rate than the furnace temperature. It will be recalled that the opposite effect obtains in a convection superheater, due to the gas flow and temperature, and consequently the heat transmission, increasing at a higher rate than the steam flow. This has led some designers to use a combination of radiant heat and convection superheaters in series with a view to obtaining a flatter characteristic curve.

The underlying principle is indicated in Fig. 116, which shows approximate typical individual curves for these superheater types, and the curve resulting from combined operation in series. The combined curve is much flatter.

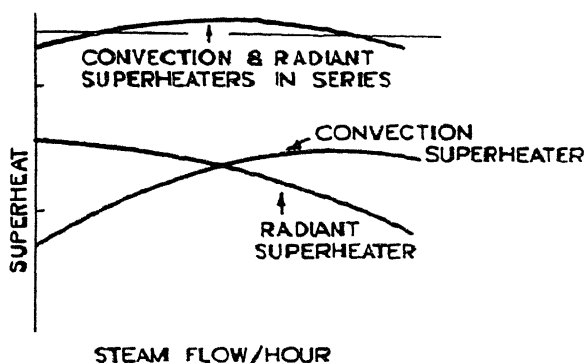


FIG. 116. Typical performance curves for convection and radiation superheaters, and for a combination of the two.

Other designers have preferred to adopt a form of "controlled" superheater whereby a primary and a secondary superheater are coupled in series, a desuperheating appliance being arranged in the steam flow between them. This method allows for changes in the rate of steam flow, the cleanliness of the heat-absorbing surfaces, etc., and results in a constant final steam temperature independent of all other operational variables.

Yet another method of superheat control is to provide by-pass passages and suitable dampers so that the gases can be diverted when the steam temperature is too high. The dampers may be manually or automatically operated.

## ECONOMISERS

An economiser, at one time, was regarded as a heat-recovery appliance placed in the path of the flue gases to abstract some of the heat that would otherwise be rejected to the chimney. Nowadays an economiser may be regarded as an extension of the boiler heating surface added as a means of obtaining higher efficiency at less cost than if boiler heating surface were provided for the purpose.

An economiser, therefore, must be proportioned with the general plant conditions in mind if the maximum benefit is to be obtained from the minimum capital expenditure, and the benefit increases with the cost of fuel used in the boiler. It is estimated that about 1 per cent. of fuel costs can be saved for every



10° F. increase in the temperature of the boiler feed water, so that where very inefficient boiler plant is being considered, savings up to 15 and 20 per cent. can be effected.

When more heat is available than can be used in increasing the sensible heat of the feed water, it may be necessary to employ a "steaming" economiser or reject the surplus heat to the chimney, if additional air heating cannot be adopted. In the average economiser, however, the feed water is not heated higher than to within 50° to 30° F. of the temperature corresponding to the boiling point of the boiler water, thus preventing any possibility of steam generation and subsequent water-hammer risks. A useful, if unconventional use of the economiser is to provide hot water for the factory for central heating, or for offices.

*Economics of Economisers.* The economic aspect of the installation can be regarded from two points of view. It can be considered that for a fixed evaporation the economiser provides a portion of the total heat added to the feed water to convert it into steam, and, since this heat is obtained for nothing, it effects an appreciable saving in fuel. Alternatively, for a fixed weight of fuel burnt per hour, less heat is ejected to the chimney, more heat is converted into steam and more steam is generated.

The following simple examples will illustrate these two points of view.

In a Lancashire boiler plant operating at 120 lb. per square inch gauge with no superheat and being fed with water at 120° F., the total heat in 1 lb. of steam above that in the feed water is 1,193 — (120 — 32) = 1,105 B.Th.U. If an economiser is installed and heats the feed water from 120° to 270° F. it adds approximately 150 B.Th.U. to the feed water and for the same hourly evaporation effects a saving in fuel of  $\frac{150 \times 100}{1,105} = 13.6$  per cent.

Suppose there is in an electric power station a boiler generating 100,000 lb. of steam per hour and operating at 600 lb. per square inch gauge pressure, with a final temperature of superheated steam of 850° F., the feed water leaving the turbine bleeder heater and entering the boiler unit at 300° F. Suppose, moreover, that by the installation of an economiser the feed water is heated from 300° to 450° F.

	B.Th.U./lb.			
Total heat of steam at 600 lb. per square inch gauge superheated to 850° F. .. .. .	..	..	..	1,435
	B.Th.U./lb.			
Heat in feed water at 450° F. .. .. .	..	..	..	430
.. .. . 300° F. .. .. .	..	..	..	270
Heat added in economiser .. .. .	..	..	..	160
Heat in steam at 600 lb. and 850° F. with feed water at 300° F. .. .. .	..	..	..	1,165

Without an economiser all this heat must be added in the boiler and superheater. With an economiser, 160 B.Th.U. of this heat are added in the economiser, leaving 1,165 — 160, or 1,005 B.Th.U. to be added in the boiler and superheater.

Therefore the evaporation will be  $100,000 \times 1,165/1,005 = 115,900$  lb. per hour.

It is thus seen that for the same size coal-handling plant, mechanical stoker, combustion chamber, boiler, superheater, forced and induced draught fans, grit arrestor, gas-washing plant, chimney, ash-handing plant and boiler-house

building, the total evaporation has been raised by 15.9 per cent. This means that practically for the same capital expenditure 15.9 per cent. more steam is generated.

The relative size of economisers designed to reduce the flue gas temperature to 280° F. is shown in Fig. 117 (P. H. N. Ulander, *J. Inst. Fuel*, I., 188).

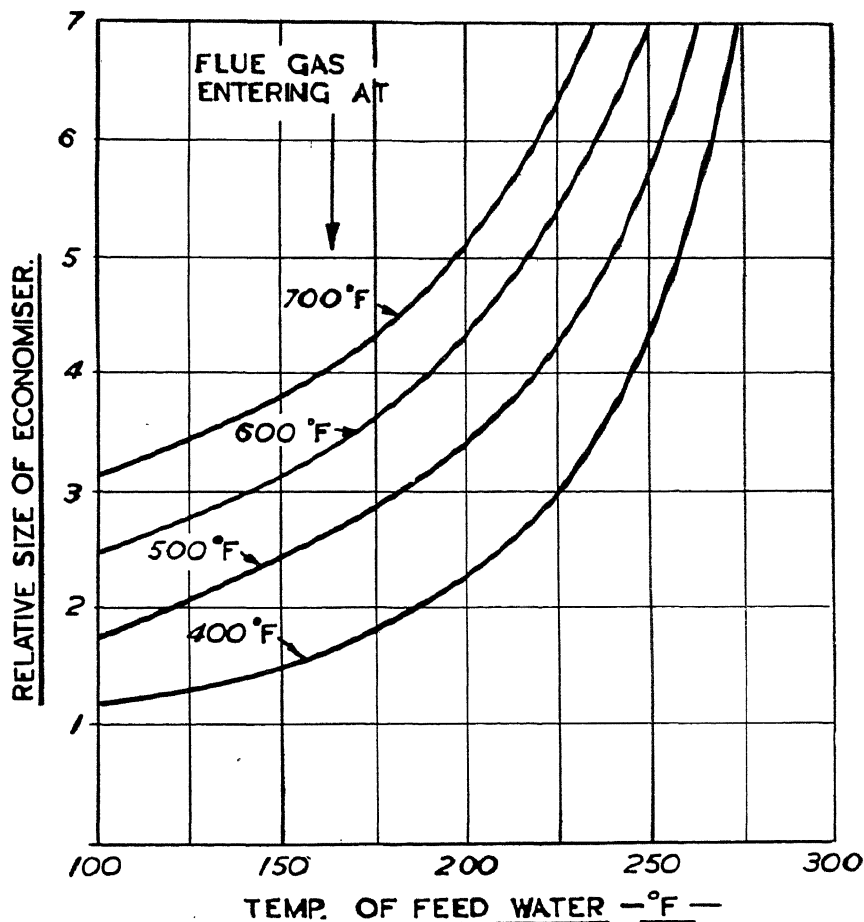


FIG. 117. Relative size of economiser to reduce flue gas to 280° F.  
(P. H. N. Ulander, *J. Inst. Fuel*, I., 188.)

There are several indirect advantages obtained by the installation of economisers, the three most important of which are as follows:—

The feeding of the boiler with water at a temperature near the boiling point reduces the temperature differences in the boiler, prevents the formation of stagnant pockets of cold water and thus reduces greatly the temperature strains created in the pressure parts of the boiler and promotes better internal circulation.

Where the feed water is not as pure as it should be, the temporary hardness is deposited on the inside of the economiser tubes, and while this necessitates

internal cleaning of the economiser, the evil is usually not as great as internal cleaning of the boiler. The temperatures experienced in the economiser not being as high as those in the boiler, the salts are sometimes deposited as a soft sludge and can be blown down through the economiser blow-down valve instead of forming a hard scale on the inside of the boiler. Even where the unwanted hardness is deposited as a hard scale on the inside of the economiser tubes the economiser can be by-passed for internal cleaning without putting the boiler out of operation. If the same feed water hardness were deposited in the boiler it would probably necessitate putting the boiler out of commission for two or more weeks for internal scaling.

The passage of the flue gases over the outside of the economiser tubes acts indirectly as a grit arrestor and a large proportion of the soot and some of the fine fly ash is deposited on the tubes and scraped off into the soot chamber, which is arranged below the bottom headers. This reduces the emission of soot and grit.

*Factors affecting Heat Absorption in Economisers.* The amount of heat absorbed by an economiser depends upon the weight of flue gases available and their initial and final temperatures; upon the economiser heating surface provided; upon the mean temperature difference between the gases and the water; and upon the overall heat transfer coefficient between the gases and the water.

The temperature of the gases after they have passed over all the boiler heating surface proper may be anything from  $550^{\circ}$  to  $1,100^{\circ}$  or  $1,200^{\circ}$  F. (i.e.  $288^{\circ}$  to  $593^{\circ}$  or  $650^{\circ}$  C.).

In a given boiler working under the conditions assumed during its design, the quantity of heat available in the flue gases is fixed. The aim of the economiser designer is to reduce the size and cost of the appliance to a practical minimum, which leaves only two variables—the mean temperature difference and the heat transfer coefficient—available for modification. The mean temperature difference is a function of the gas and water temperatures, and is at a maximum when a counterflow of gas and water can be arranged, i.e. when the coldest water enters the economiser at the region where the coldest gases leave it. Counterflow, however, is not always possible, as, for instance, in some of the older forms of economiser, and thus the designer must endeavour to improve the heat transfer coefficient.

This coefficient, in turn, is derived from three other coefficients: the coefficient for the heat flow from gas to economiser tube surface, externally; that for the flow through the tube wall; and the coefficient for the flow from the economiser internal tube surface to the water inside it. The first coefficient (gas to external surface) depends principally upon the mass velocity and the temperature of the flue gases; the second is of relatively small importance in ordinary bare-tube economisers; and the third depends upon the mass flow of the water, the internal diameter of the tubes and the mean water temperature. The water coefficient is, in fact, so much larger in relation to the gas coefficient that it is usually sufficient to select a water velocity suitable for a given water pressure drop through the economiser and ensure that turbulent flow will be maintained at the lowest expected boiler rating.

*Extended Pipe Surface.* Designers have therefore been compelled to improve economiser performance by an increase in the gas coefficient, and this can be readily augmented by raising the gas velocity. The usefulness of this method, however, is restricted by the permissible cost of the extra fan power required, although it can be extended to the limit by the addition of extra heat-absorbing surface, in the form of gills, etc., to the conventional form of bare economiser tube usually fitted.

The bare tube has disadvantages in that whether it is arranged in plain or

staggered formation there are liable to be "dead" spaces behind the tube in the direction of the gas flow, and these spaces are untouched by the gases. Moreover, a certain amount of stratification is likely to occur (cf. Chapter IX), as a result of which layers of gas can pass right through the economiser without impinging upon more than a small proportion of the tubular heating surface.

The addition of extended surface to the bare tubes thus offers more surface for contact and increases the "scrubbing" action, but too much may defeat the purpose of the designer by causing an undue reduction in the temperature head available at the tube wall. It is in such circumstances that the second coefficient mentioned above—flow through the tube wall—assumes greater importance in the computation of the overall heat transfer coefficient.

*Economiser Types.* Turning now to a consideration of economiser types in general use, it will be found that the plain-tube type is widely favoured in Lancashire boiler installations working under natural draught. The tubes are made of cast iron to resist the corrosive action of the flue gases, and their ends are pressed into top and bottom headers. The method of construction restricts the application of this type to pressures below 250 lb. per square inch, but an improved construction in which a number of the tubes in each section is positively secured to the headers by bolts allows of its use up to about 550 lb. per square inch.

Plain cast iron tubes are used where space is not a restriction. The nest of economiser tubes is then arranged with the axes of the tubes vertical. Four, six, eight, ten or twelve tubes, machined at the ends and pressed hydraulically into top and bottom headers, are formed into a section. The sections are assembled in groups, the number varying according to the evaporation of the boiler plant and the duty required from the economiser. The standard lengths of tubes are 9, 11½ and 13 feet.

The bare-tube design has the inestimable advantage that the external surfaces can be continuously and directly cleaned by circular scrapers moving up and down the tubes, but if external corrosion is to be prevented, the entering feed water must be *at all times* at a temperature of at least 110° F. (43° C.) to eliminate "sweating." If corrosion is found to occur under these conditions the water inlet temperature should be raised by 20° to 130° F.

*Gilled Tubes.* A reduction in economiser size together with increased heat transmission can be obtained by casting circular gills on to the bare-tube walls, but at the higher pressures the cast iron tube is unsuitable. Steel tubes are then used, and have cast iron gilled sleeves shrunk on to them, the inner tube serving to withstand the pressure and the outer tube resisting any corrosive attack. Suitable spigot joints between the sleeve ends protect the inner tubes from external corrosion.

The ideal arrangement is for the gases to pass vertically downwards over such a group and the water vertically upwards, thus obtaining approximately contraflow heat exchange. The number of tubes and rows and their length is determined mainly by the total heating surface required and the draught loss available. It is usual to connect the tubes so that the water flows through all the tubes of the bottom row in parallel and then returns through all the tubes of the second row; then back through the third row, and so on. Thus the flow through the tubes of any one row is in parallel and the flow through the rows is in series. However, this arrangement can be changed to meet special circumstances, e.g. where a very low hydraulic loss through the economiser is desired, in which case the water passes through two rows in parallel.

Trouble with soot deposits, however, has necessitated modifications of the circular type of gill, and there are now available several forms of rectangular gills with "streamlined" surfaces designed to reduce draught losses to the

minimum consistent with high heat transfer. It will be appreciated that tubes of the extended-surface type cannot be cleaned by mechanical scrapers while in service, and must, therefore, be equipped with steam or compressed-air soot blowers. The blowers are operated at regular intervals and collect the soot in places convenient for removal.

Where weight is a major consideration as in marine economisers, an alloy of aluminium is used for the construction of the gills. Since the coefficient of expansion of aluminium is greater than that of mild steel, each aluminium gill is kept in intimate contact with the steel tube by the insertion of a ring of special steel with a coefficient of expansion less than that of mild steel.

### FEED WATER HEATERS

The installation of an economiser cannot, of course, be justified in every class of steam-generating plant, as, for instance, in a moderate-sized laundry having an Economic boiler for all purposes. The use of feed water at as high a temperature as possible is obviously desirable in such circumstances, and some means of obtaining it must be arranged.

In small factories which utilise steam for heating and similar purposes, every effort should be made to return the hot condensate to the feed tank (see Chapter XXIII). If the condensate temperature is 180° F., 1 gallon of condensate carries 1,480 B.Th.U. Thus if the fuel burned in the boiler has a calorific value of 13,000 B.Th.U. per lb. and the boiler efficiency is 65 per cent., every gallon of condensate returned to the feed tank represents a saving of approximately one-seventh of a lb. of coal based on feed water at 60° F. In addition, the water so returned is free from the scale-forming compounds found in the raw water, thus assisting in reducing boiler cleaning charges.

Hot feed water can be obtained by blowing live steam into the feed tank through one of the "silent" jets now available, but the real economy of this practice is somewhat doubtful. When a factory develops all or a part of its power by means of a steam engine, feed water heating can be effected by discharging sufficient of the exhaust steam into the tank. In a multi-expansion engine steam can be tapped off between the expansions, but in either arrangement provision should be made so that a vacuum cannot form in the steam pipe and cause feed water to be drawn into the engine cylinder.

If the engine exhaust contains oil the open exhaust method cannot be used, and an oil separator must be incorporated in the system. If adequate volumes of exhaust steam are available, it may be advisable to install a feed heater of the direct-contact type which not only heats, but also de-aerates the water. Heaters of this kind are arranged on the suction side of the feed pump, at the requisite height, and as they are open to the atmosphere the maximum feed temperature obtainable is limited to 212° F.

### ENCLOSED HEATERS

Higher feed temperatures up to that of the boiler water are possible by using one or more heaters of the enclosed type. These invariably assume the form of a closed body which houses tubular heating surface, the heating steam flowing over the external surfaces of the tubes while the water is pumped through them.

For the lower pressure and temperature ranges the tubes may be secured in two tube plates, as in the normal surface condenser, but for the higher ranges the relative expansion of tubes and body prohibits this construction. Various types of "floating" headers, U-tubes, etc., have been evolved to meet the need, and the principal differences between makes of feed heaters are in that respect.

Considerable use of surface heaters has been made in connection with turbo-generator sets, the heating steam being "bled" from the machine after the

steam has done a certain amount of useful work. To obtain the theoretical optimum efficiency, an infinite number of heaters should be employed, but in practice the number has so far been confined to five or six. Steam which has passed the saturation point is generally used, and when more than one heater is justifiable the remaining adiabatic heat drop may be suitably subdivided from this point.

If a high pumping loss is to be avoided, the velocity of water through the

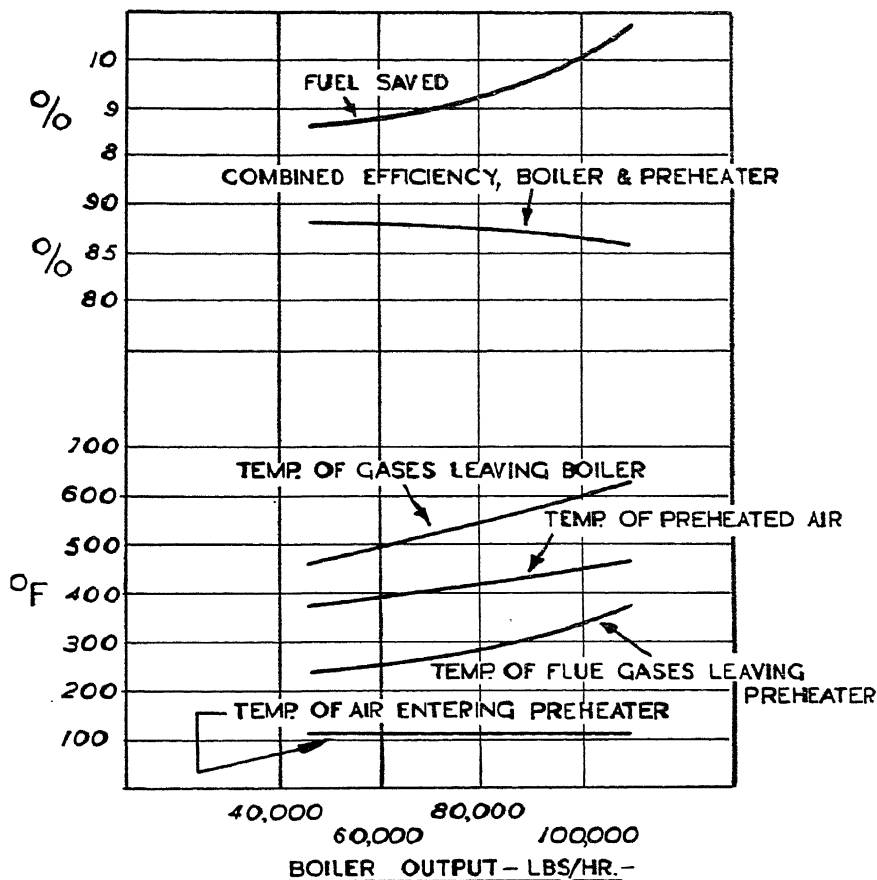


FIG. 118. Test results showing fuel saved by air-heater without economiser.  
(P. H. N. Ulander, *J. Inst. Fuel*, I, 188.)

heater tubes is important, and this usually varies from about 4 to 7 feet per second. The size and cost of the heater can be reduced by increasing the water velocity, and, consequently, the heat transfer coefficient, but moderate velocities are desirable for the reason given. The tubes vary in external diameter from  $\frac{1}{2}$  to 1 inch, and the number required to give the necessary cross-sectional area is directly related to the water velocity adopted.

The tubular surface area (and the tube length, when the diameter is determined), depends upon the weight of water to be heated; its temperature rise; the heat transfer coefficient; and the mean temperature difference between the

steam and water. The number of water passes required in the heater will vary according to the permissible height or length of the heater, but when the two-tube-plate construction is used, the length between tube plates should not be greater than about 100 tube diameters, or tube-supporting plates may be necessary. These, it will be appreciated, interfere with the free flow of steam internally.

## AIR PREHEATERS

The air preheater is an important auxiliary in modern boiler installations, and transfers heat from the flue gases to the air fed to the furnace for combustion purposes.

The effects claimed from air preheating are as follows :—

- (1) The recovery of heat from the flue gases reduces the heat loss from this source. It is claimed that every 30° to 35° F. reduction of the flue gas temperature in this way results in a fuel saving of  $1\frac{1}{4}$ – $1\frac{1}{2}$  per cent.
- (2) The flame temperature in the combustion space is increased with consequent greater rate of heat transfer by radiation (cf. Chapter VIII), thus increasing the effective heat transfer and increasing the amount of steam made per boiler.
- (3) With higher flame temperature, the fuel can be burned with less excess air. In addition to the better utilisation of heat (Chapter V) there is also from this cause a reduction in the power expended in producing draught.
- (4) Frequently lower grades of fuel, or different grades, can be burned successfully with preheated air, which would not be properly burned with cold air. Some coals can be burned satisfactorily only with preheated air.

Fig. 118 (Ulander, loc. cit.) indicates from test results the advantageous effect on fuel consumption of using preheated air.

*Types of Air Preheater.* Preheaters are either regenerative or recuperative, and in general principle are similar to the regenerators and recuperators used in furnace work (Chapter XVIII).

With recuperative heaters the gases pass on one side and the air on the other side of the tube or plate wall, the heat transfer taking place through the plate. All tubular and plate heaters come within this group.

Recuperative heaters are more economic on small installations and where a small proportion of the available heat has to be recovered, but deposits on the plates affect the rate of heat transfer.

Regenerative heaters operate by passing hot flue gases and cold air alternately through the heating elements. Heat is picked up by the elements from the gases and in the next stage of the cycle is transferred to the air. Only one heater of this type is marketed in this country for use with steam boilers. In this apparatus, the element plates are assembled in a cylindrical housing, which is rotated slowly so that the elements pass alternately through the flue gases and air during each revolution. The elements are heated by the gases and then cooled by the air, thus the heat transfer is continuous.

*Temperature Limits.* With each method of burning fuel in a boiler there is a limit to the temperature of the combustion air, and if this limit is exceeded the maintenance charges on stokers, burners or furnace parts increase rapidly. The maximum hot-air temperatures are approximately :—

Method of firing		Temperature F. °C.	
Hand. . . . .	.. .. .	250–300	121–149
Sprinkler and chain-grate stokers	.. .. .	250–300	121–149
Retort stokers	.. .. .	450–500	232–260
Pulverised coal or oil firing	.. .. .	Up to 600	Up to 315

On the gas side of an air heater the maximum entering temperature is restricted to 750°–800° F. (399°–427° C.), which is the temperature that mild steel will safely withstand should the air flow cease. Beyond this limit it is necessary to introduce special designs or special operating precautions, or to make the heater elements of high-temperature resisting alloy steels, thus considerably increasing the cost.

Many air heater troubles result from working with too low an exit gas temperature. The corrosion difficulty has been explained when dealing with economisers. In addition, the deposition of moisture will cause dust to stick to the heating surfaces, which impairs heat transmission. The prevention of condensation within an air heater is therefore imperative.

*The Prevention of Condensation.* With recuperative heaters the most satisfactory method of maintaining the requisite element plate temperature at the air inlet end is to return some of the heated air to the forced draught fan inlet. Thus the temperature is raised to just above that at which condensation would occur in the flue gases. As the temperature of the element plates cannot fall below the entering air temperature, this method has successfully overcome corrosion and deposit troubles. An amount equivalent to 30 or 40 per cent., or even more, of the air passing through the heater may have to be re-circulated in this manner to ensure that the air temperature at the heater inlet is at the required figure, usually between 115° and 130° F., according to fuel and combustion conditions.

The makers of the regenerative heater claim that re-circulation of air is not necessary in their design. Because of its rotation and of the absence of sections with a cross-flow of gas and air, cool spots caused by uneven air and gas distribution at the air inlet end do not occur, and all the element plates are subjected to the same minimum temperature conditions.

In boilers of the Lancashire, Economic, marine Scotch types, etc., the gas temperature at the heater outlet should not at any time during operation fall below 280°–300° F. (138°–149° C.). In water-tube boilers, which are usually larger and obtain their air supply from above the boilers where the air temperature is already high, combustion conditions are in general steadier and more closely controlled, so that gas temperatures down to 240° or 230° F. (115°–110° C.) may be permitted when the fuel and combustion conditions are favourable.

The most satisfactory means of controlling the gas outlet temperature above the required limit at reduced loads is to provide air by-pass ducts and dampers. By opening the air by-passes, some air is short-circuited past the heater, less heat is taken from the flue gases and the gases then leave the heater at a higher temperature. The heater elements are thus maintained at the proper temperature.

For satisfactory control of an air heater it is usually necessary to provide a gas by-pass having a tightly shutting damper, isolating dampers to prevent gas flow through the heater during starting-up, banked-fire and shutting-down conditions, and air by-pass ducts with regulating dampers. On recuperative heaters, air re-circulating ducts and regulating dampers are advisable. A forced draught fan should be installed. Many air heaters are designed for a low draught loss on the gas side so that they can work in conjunction with chimney draught only, but it is usually found advantageous to fit an induced draught fan.

## AUXILIARY POWER AND STEAM UTILISATION

The efficient generation of steam is of primary importance if fuel supplies are to be conserved, but the utilisation of the steam produced is also of equal importance. Unfortunately, the average works tradition is such that the



steam-plant engineer can rarely exercise any influential control over the production equipment in which the steam is used, so that boiler and power-house economies are liable to be neutralised by wasteful production methods. The plant engineer can, however, control the fuel-consuming appliances required to operate his auxiliary boiler and power-house plant, and brief mention will be made of some of the aspects of the subject that need consideration to achieve the best results.

Most modern auxiliaries are either steam or electrically driven, and the final decision of selection will depend largely upon individual conditions. The advantages of a steam drive are that steam plant will stand a certain amount of abuse before complete failure supervenes ; it is useful as a stand-by in the event of an electric power failure ; it can be used for starting up the plant when no electric power is available until the main generator is on load ; and the exhaust steam can be used for feed heating or for process work.

Steam plant has, on the other hand, usually high initial, maintenance, foundation and lubrication costs. Its thermal efficiency is low (unless the exhaust steam can be utilised), and it occupies a comparatively large floor area. Steam plant cannot be readily adapted for remote control, thus preventing the provision of centralised controls, and there are the inevitable losses and leakages which necessitate extra feed water make-up.

An electrical drive has the advantage of a good speed regulation, and it lends itself to centralised control, either automatic or remote. It is very flexible, and the initial, foundation and lubrication costs are low. Moreover, the floor space required is relatively small. Electrical drives are, of course, useless unless electrical power is available for starting-up purposes.

When electrical energy is privately generated for process work by a straight condensing turbine or reciprocating engine having no bleed-steam connections, the provision of steam-driven auxiliaries exhausting to feed water heaters and evaporators, etc., will raise the feed temperature and thus increase the overall thermal efficiency of the installation. When both steam and power are generated for process work, the choice of drive frequently depends upon whether there is a surplus of steam or of electrical power. If there is normally a good balance, then the capacity of the main generator to meet auxiliary loads must be taken into consideration. In this respect, it must be remembered that a main turbine or reciprocating-engine generator has a higher efficiency than any small turbine or engine. Again, when steam is generated for process purposes only—all electrical power being purchased outside—the installation of back-pressure turbines and electrically driven auxiliaries may reduce power costs.

In the ordinary boiler house, power is required for operating mechanical stokers, fans, coal and ash elevators, feed pumps, etc., while steam may be required for feed water heating and occasionally to ensure efficient combustion. To these must be added the steam necessary to operate air ejectors, soot blowers and similar appliances.

Generally speaking, the equivalent steam consumption for boiler house auxiliary purposes in industrial plants should not greatly exceed about 10 per cent. of the total amount of steam generated, but if it does, an investigation into the cause of the excess is indicated. Mechanical stokers should not take more than 0.3–0.5 per cent. of the total steam, or its equivalent, and the coal and ash elevators may require from 0.8 to 1.0 per cent. The boiler feed pumps, on the other hand, may use from 2.5 to 3.5 per cent. of the total steam generated.

When steam is required for feed water heating it can usually be obtained by bleeding from the main prime mover, or by using the exhaust from the auxiliaries, but steam for forced draught steam jets as used on Lancashire and similar boilers must be taken from the high pressure line, and may use from 3 to 4 per cent. of the total steam generated.

## BOILER FEED PUMPS

Boiler pumps fall into two broad classes :—

- (1) Displacement pumps, in which the feed water is displaced from the suction to the discharge by the reciprocating motion of a bucket or plunger.
- (2) Centrifugal pumps, in which the feed water flows through the pump by reason of the centrifugal force imparted to the water by the rotation of one or more impellers through which the liquid flows from the suction to the discharge.

Direct-acting feed pumps of the reciprocating type are displacement pumps ; they are by far the most common type of feed pump in use.

Centrifugal feed pumps are built for capacities up to 500 tons per hour, and are usually provided in the larger power plants.

A little consideration will show that the first requirement of a boiler feed pump is reliability. Operating, maintenance and first costs are important, but fade into insignificance when a pump ceases to function at the time of a heavy steam demand on the boiler or boilers that it serves. Due to the designs of a number of leading manufacturers, the direct-acting type of feed pump has a well established reputation for reliability in Great Britain, and these and other makers have built up a similar reputation for the centrifugal type of boiler feed pump now used in many industrial and power-station plants.

The direct-acting pump is designed for duties up to 200,000 lb. of water per hour, but is more generally applied as a unit to boilers totalling about one-half this capacity. The size of pump required for particular conditions will, of course, depend upon the maximum evaporative capacity of the boiler or boilers, due allowance being made for the pressure and temperature conditions obtaining. The direct-acting pump can be constructed as a single- or two-cylinder unit, vertical or horizontal, depending upon the purpose for which it is required. The vertical simplex type saves floor space and has only one steam and one water end to keep in order, while the horizontal type may cost slightly less.

The principal differences between the various designs of pumps apart from those just mentioned lie largely in the steam-distribution mechanism and the amount of expansive working thereby rendered possible. Normally the cut-off is not less than about 0.7 of the piston stroke. When the exhaust from the pump can be utilised in a feed heater, however, a lower steam consumption may be partly neutralised if a more intricate valve gear, necessitating frequent maintenance, is required. In such conditions it may be advisable to install a more rugged, if less economical form of pump.

Pump speeds vary from 50. to 80 feet per minute, but generally the slower pumps suffer the least wear and tear, and therefore have lower maintenance costs. Bearing surfaces, particularly the moving parts of steam valves, should be ample, and full provision should be made to allow of the rapid overhaul of steam and water ends.

The direct-acting pump is probably the best appliance for boiler feed purposes for pressures up to about 200 lb. per square inch pressure. In plants using high steam pressures and high superheat temperatures a reciprocating form of pump can still be employed, but in this instance a motor drive must be adopted. The pump is usually of the multi-throw type, driven through reduction gearing, and the discharge pressure varies almost directly as the output. As will be shown later, the centrifugal pump installed in many modern boiler houses can set up a high pressure differential when the feed check is practically closed. The efficiency curve of a reciprocating pump is reasonably uniform over the

range of its capacity, and it has, in fact, been installed to serve some of the high-pressure boilers now in service.

*Centrifugal Boiler Feed Pumps.* The direct-acting pump, as mentioned above, is constructed for duties up to 200,000 lb. of water per hour, but, according to individual circumstances, its economical upper limit is in the region of 150,000 lb. per hour. The lower economical limit of the centrifugal boiler feed pump is about 75,000 lb. per hour, so that a pump purchaser in the 75,000–150,000 lb. per hour range must decide on the respective merits of the

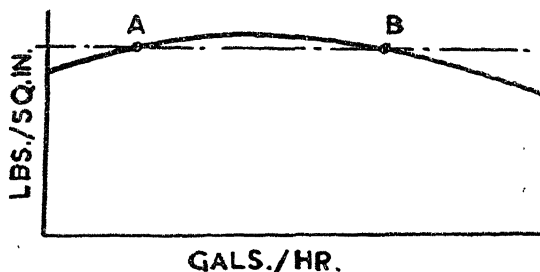


FIG. 119. Relation between pressure and output for a centrifugal pump.

two types. In such circumstances the best solution is probably to consult a manufacturer who produces both direct-acting and centrifugal boiler feed pumps, and carefully to consider the data thus obtained. In general, the centrifugal pump gives satisfactory service when its output is relatively large compared with its pressure.

The pressure developed by a centrifugal pump varies roughly as the square of its peripheral speed, and the blade angle at the circumference of the impeller. With an ordinary type of centrifugal pump, the pressure rises at first and then falls with increased output, so that a curve of the form shown in Fig. 119

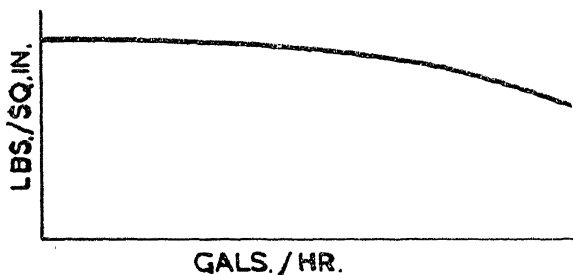


FIG. 120. Characteristic curve of a centrifugal pump.

results when pressure is plotted against capacity at constant speed. Obviously, a pump working under such conditions is liable to oscillate between two rates of delivery, A and B, and each change from one point to the other causes a pressure impulse to be set up in the pipe-line.

This unstable method of operation is unsuitable for boiler feed lines because of the danger of water hammer, and it is therefore necessary to use a pump that works on the "falling" part of the characteristic curve, as indicated in Fig. 120. This is particularly important when two or more centrifugal pumps are to serve in parallel, otherwise there is the possibility that one pump will give its maximum discharge while the others are practically idling and heating

up. The slope of a characteristic curve should not, however, be too steep, or there will be a large pressure difference between full-load and no-load conditions. The permissible pressure increase may vary from about 8 per cent. with a constant-speed electrical drive to perhaps 10 per cent., or slightly more, when a steam-turbine drive is used. Too large a pressure increase reacts on the pressure-bearing parts of appliances such as economisers, feed heaters, etc., located between the feed pump and the boiler.

*Centrifugal Pump Construction.* Centrifugal pumps can be designed to discharge into a volute surrounding the impeller, and in this arrangement the kinetic energy of the water is converted into pressure energy inside the casing itself. The method is unsatisfactory in that heavy bending stresses may be imposed on the impeller shaft, due to unequal radial thrust, and most designers of modern feed pumps provide a ring of diffusers around the impeller periphery. The diffusers gradually convert the kinetic into pressure energy, and also serve to guide the flow when a multi-stage pump is used.

For a single-impeller centrifugal pump to operate at the usual boiler pressures, the impeller speed and diameter must be large, and this may result in restricted water passages through the impeller. Again, the frictional losses of an impeller vary approximately as the cube of the diameter and the square of the velocity, so that a considerable reduction in losses can be achieved by using two or more comparatively small impellers working at a lower speed as against one large impeller working at a high speed.

The multi-stage construction has the further advantage that the subdivision of the total increase in pressure lessens the impeller clearance losses, and is now largely used for boiler feed pump purposes.

*Selection of Pumps.* Other points to be considered when analysing the construction of a centrifugal feed pump are the method of allowing for axial thrust, the prevention of leakage where the pump shaft projects through the casing, the provision for expansion at high temperatures, and the means adopted to minimise overheating of the pump under closed feed valve conditions.

One method of reducing the effect of axial thrust is to arrange the impellers so that the respective thrusts tend to cancel out, but this method usually also requires the provision of a thrust bearing to absorb any residual forces. Perhaps the most universally adopted construction is the balancing disc or drum, which, due to the water pressure on its face, opposes the axial thrust on the impellers.

The arrangements for preventing leakage between the pump shaft and its casing range from the simple packed gland to elaborate water-cooled devices. Some designers incorporate a form of labyrinth gland in the balancing drum which reduces considerably the pressure and temperature to which the outer glands are subjected, while others rely on separate labyrinth glands and water or air cooling.

In pumps for very high pressures and temperatures, it may be necessary to provide an auxiliary centrifugal pump solely for water-cooling purposes. Radial expansion of the pump can be largely counteracted by adopting centre-line suspension and longitudinal expansion by fixing one end and leaving the other free to move, but the merits of a vertical method of suspension should not be overlooked in this respect. For very high temperatures, at least one maker completely water-jackets his pump. Overheating under closed feed valve conditions is usually reduced by providing a leak-off valve, manually or automatically controlled.

## PUMPING HOT FEED WATER

The saving by using hot condensate as boiler feed water is two-fold, firstly because its use means a direct economy in fuel, and secondly because its purity will do much to prevent scale in the boiler feed system. It is found, however,

that valuable condensate is often wasted because of the difficulties of pumping hot water, and this means that tons of fuel represented in the hot condensate are run down the drain while more fuel is used in heating cold feed water. While it is agreed that there are theoretical limits and practical limits in handling hot condensate, a very considerable economy can be effected if the necessary consideration is given to the design of the pump, suction piping and position of feed tank. Experience has shown that the source of the trouble is usually the suction piping or the position of the feed tank.

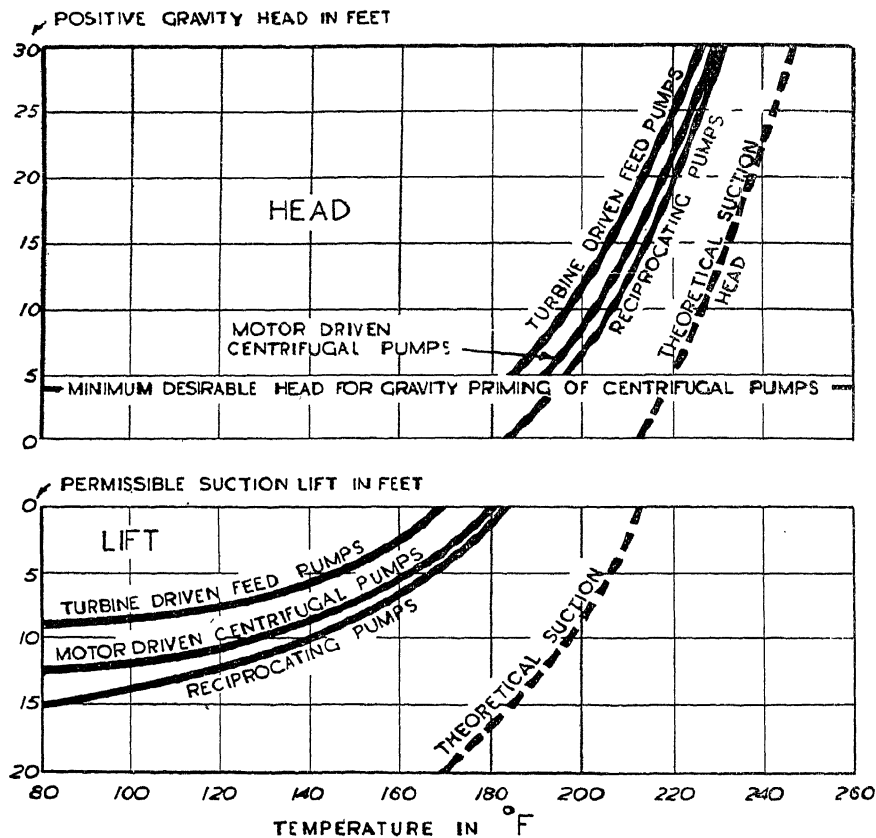


FIG. 121. Suction heads allowed for standard designs of pumps.

The secret of successfully pumping hot feed water is attention to the correct conditions on the suction side of the pump. Flow from a pump is caused by the pump, but the flow of a liquid into a pump is caused by forces outside the pump. A pump should, therefore, be so arranged that, from the source of feed water supply, a pressure is created in the suction chamber of the pump greater than the pressure at which the water will vaporise. When water is being pumped from a tank containing a free surface, the difference between the pressure on the surface of the water and the pressure in the suction chamber of the pump is the only pressure available to cause the flow of liquid into the pump. Fig. 121 shows the usual pressure difference allowed (expressed in feet of water)

## STEAM BOILER AUXILIARY PLANT

for the average installation of various types of pump of standard design. The dotted curve shows the boiling point of water. The pressure difference allowed includes a reasonable allowance for the friction losses in the suction pipe line and suction passages and valves in the pump, and includes a margin to cause the liquid to follow the motion of the moving parts of the pump without vaporisation occurring.

Generally speaking, suction heads require to be greater for high speed or large capacities than for low speed or small capacities.

Before water can flow into a pump, the air or vapour in the suction line must be evacuated sufficiently to reduce the pressure in the pump to cause the liquid to flow into the suction chamber. Displacement pumps do this automatically when they are started. In centrifugal pumps, the liquid must be present in the first impeller before the centrifugal force to cause flow can be imparted to the liquid, and this requirement necessitates that the pump be placed relatively to the source of water supply so that the water flows by gravity into the impeller. If this cannot be done, special priming devices must be used to cause the evacuation of the air and flood the pump with water so that the motion of the pump can create the centrifugal force to cause a flow from the pump which will establish the flow from the feed tank into the pump.

A centrifugal boiler feed pump should never be arranged to have a suction lift unless the site conditions are such that it is impossible to arrange the pump and the feed tank to give a gravity flow into the pump.

Table 67 gives for direct-acting feed pumps the information in tabular form that is also in Fig. 121.

The suction piping from the feed tank to the pump should be as short and as free from bends as possible. Sharp right-angled bends should certainly be avoided, and the suction piping and valves should be designed to reduce the friction losses to a minimum.

It will be seen, from the figures given, that with the temperature above 175° F., the feed tank must be placed higher than the feed pump to ensure a pressure at the pump suction. The head required between the pump suction and the water level in the tank must be sufficient to prevent the water vaporising in the pump end.

TABLE 67. SUCTION LIFT AND HEAD FOR VARIOUS FEED TEMPERATURES.  
DIRECT-ACTING FEED PUMPS

Temperature ° F.	Lift in feet to suction valves
130	10
150	7
170	2
175	0
	Head over suction valves in feet
190	5
200	10
210	15
212	17

For feed temperatures *over* 212° F., it is necessary to provide a suction head of 17 feet plus a head equal to the gauge pressure corresponding to the feed temperature. If the feed temperature is over 212° F. and the heating is all carried out in the feed tank, the tank must be sealed and elevated at least 17 feet above the feed pump, or a booster pump provided to give the necessary suction head. Alternatively the feed heating can be carried out in a direct

contact or a surface heater and suitable booster pumps provided to give the necessary pressure at the suction of the feed pump.

These briefly are the principal points requiring consideration in pumping hot condensate, but if they are carefully considered and the plant correctly arranged, there should be no difficulty in handling hot water, with increased plant efficiency and a considerable saving in fuel.

## MAINTENANCE OF FEED PUMPS

In the maintenance of feed pumps, there are two broad principles to be followed :—

- (1) The various component parts of the pump should be maintained in good condition so as to reduce to a minimum all possible leakages of water from the discharge of the pump to the suction of the pump or to any intermediate pressure within the pump. All discharges of water from the pump to the atmosphere should also be reduced to a minimum. Any leakage of water from a place of high pressure to another place at a lower pressure represents a loss in pumping energy, and such loss should be reduced to a minimum by maintaining the pump in an efficient condition.
- (2) Similarly, every care should be taken to ensure that the steam end of a steam-driven pump is maintained in the best possible condition so as to reduce to a minimum all possible leakages of steam from the steam supply pressure to the exhaust pressure or to the atmosphere. Such leakages represent a loss of heat energy and, where the steam escapes to the atmosphere, there is also a loss of distilled feed water.

## UTILISING EXHAUST STEAM

The subject of exhaust steam utilisation is discussed in detail in Chapter XXIII. Here reference is made only to its use for heating feed water.

The theoretical and practical limits in handling heated feed water have been illustrated and described in relation to Fig. 121, and all practicable steps should be taken to ensure that the feed water which the feed pump is required to handle should be at the maximum possible temperature which the pump is capable of dealing with. For this purpose, consideration should be given to the position of the feed tank and the design and layout of the suction piping to enable the pump to operate with the maximum possible feed water temperature.

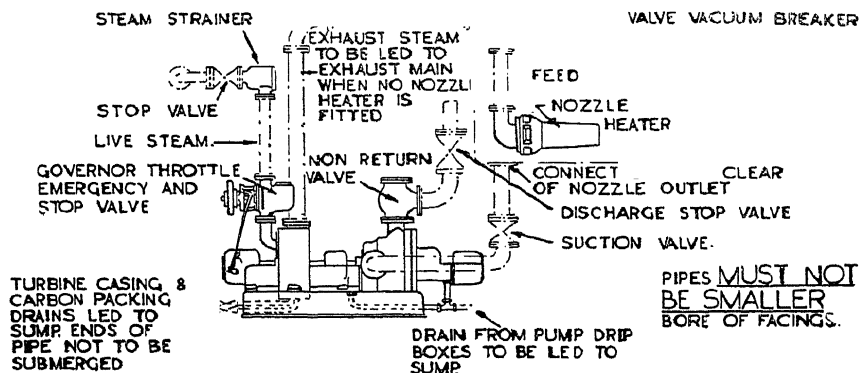


FIG. 122. Typical arrangement of turbine-driven feed pump exhausting to nozzle heater in feed tank.

The simplest method of using exhaust steam is to discharge it into the feed tank through a mixing nozzle. The exhaust steam from the turbo feed pump is not contaminated with oil and can, therefore, be condensed by direct contact with the feed water in the feed tank, thereby conserving the whole of the heat in the exhaust steam and also the distilled water formed by the condensation of the exhaust steam. A convenient arrangement for a turbo-driven feed pump is illustrated diagrammatically in Fig. 122. The quantity of exhaust steam to be dealt with is considerable and the guidance of feed pump manufacturers should be sought to ensure that the best possible arrangement of satisfactory heating is obtained and that the arrangement adopted will be entirely satisfactory so far as the operation of the feed pump is concerned. It may be that the arrangement of the plant and the operating conditions are such that it would be preferable to provide a surface feed heater on the discharge side of the feed pump into which the turbine of the feed pump discharges the exhaust steam for feed heating purposes, the feed water being discharged by the pump through the tubes.

### DRAUGHT PRODUCTION

The rate at which coal can be burned in a boiler furnace per square foot of grate area depends, among other things, upon the characteristics of the fuel being burned: a non-caking coal, for instance, offers less resistance to air flow than does a caking coal, and thus absorbs less of the available air-pressure head. Again, some coals of the anthracite class will hardly burn at all unless provided with a high draught intensity, while others require only a low draught, or disproportionate fly-ash losses result. For every class of coal, therefore, there is a rate of burning which gives the most efficient combustion conditions, and the means of providing draught must be selected with this in mind.

*Chimneys.* The most common method of supplying air to boiler furnaces is by means of a chimney, and this has been discussed in some detail in Chapter VI. The height and diameter of a chimney must be determined from a consideration of present and future draught requirements, so that in a business likely to expand rapidly, it may be financially advisable to provide a larger chimney than can be justified on immediate needs.

*Steam Jets.* A chimney, however, is a very inefficient appliance for the production of draught, depending, as it does, upon a high gas temperature at the boiler flue outlet. Moreover, the flexibility of a chimney is limited. In such circumstances, a boiler operator may adopt some system of artificially-produced draught, the most common being that in which steam jets are fitted at the chimney base or below the firebars. The first method is used principally in vertical or locomotive boilers, and causes an induced draught which permits a higher rate of combustion or a reduction in the height of chimney required to produce a given draught. The second method is widely used in horizontal internal-flue boilers and in conjunction with various types of mechanical stokers fitted to such boilers. It causes a forced draught of air through the fuel bed, and consequently lessens the draught required to draw off the products of combustion.

Steam jet draught also assists in cooling the firebars and influences clinker formation and, in fact, is essential with certain classes of coal if combustion is to be ensured. When new, these jets serve the purpose and are at present fitted to many boilers having to burn inferior coals.

Steam jets should not consume more than 3-4 per cent. of the total steam generated by the boiler to which they are fitted, but unfortunately the jet orifices are liable to be considerably enlarged by erosive action. The greatly increased steam consumption that may arise from wear of steam jets has been indicated in earlier chapters.



*Fan Draught.* Artificial draught can be produced at a higher capital cost, but with greater economy and efficiency by installing a fan system. A great deal of the improved performance claimed for the modern internally-fired multitubular boiler can be attributed to the use of higher gas velocities, which the boiler operator must, of course, pay for in fan power. A saving in the height, and consequently cost, of a chimney is effected, but it must not be overlooked that most local authorities stipulate a minimum chimney height.

The use of fan draught allows the maximum amount of heat to be abstracted from the products of combustion before they are discharged into the chimney; it makes possible enhanced rates of combustion of low-grade fuels together with higher furnace temperatures and rates of convective heat transfer; and it permits rapid control of the boiler over a wide range of output. Fan draught can be applied as forced, induced or balanced draught, according to particular boiler requirements; the first having the advantage that any air leakage is outwards; the second that it probably gives a better distribution of air through the fuel bed; and the third that it lessens the liability to cold air leakages into the furnace.

*Fan Types.* In considering the type of fan to be adopted for boiler draught

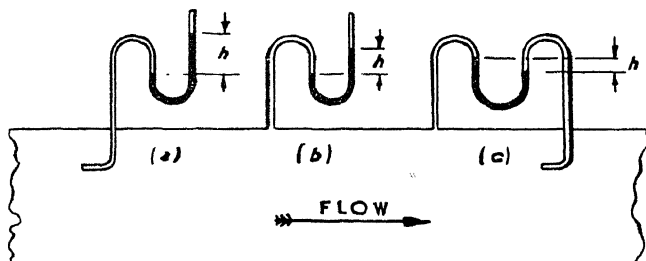


FIG. 123. Static and velocity heads at fan.

purposes it is necessary to enquire into the kind of pressure required. The total pressure produced by a fan is represented by the manometer at (a) in Fig. 123, and is the pressure due to the sum of the static and velocity heads. The static head can be indicated by a manometer of the type shown at (b), but to obtain the velocity head it is necessary to insert a tube into and facing the main stream of air and to connect its other end to the wall of the ducting, as at (c) in the figure (cf. Chapter IX). In boiler work it is the static head that has greatest importance, since it is by virtue of static pressure that the resistance to flow through, say, a compacted fuel bed, can be overcome.

The static head varies according to the fan blade curvature. Blades having a forward curvature, or those of the straight radial type, develop higher absolute air velocities than do those of the full backward-curved type, and they are less efficient. On the other hand, the backward-curved fan requires a higher peripheral speed for a given static pressure, and must therefore be of heavier build.

The resistance-overcoming properties of the backward-curved fan make it very suitable for forced draught work, but its necessarily heavier construction militates against its use for induced draught production. In the latter application there is a tendency for deposits to collect on the reverse faces of the blades, with obvious results on the fan balance, and extra thickness must be provided to allow for abrasion of the blades in service. A backward-curved blade would have to be of heavy section, as stated above, and it is therefore principally used for forced draught service, the radial self-cleaning form of blade being adopted for induced draught requirements. It will be appreciated, of course, that there

are several intermediate types of fan blades used for various boiler house purposes.

*Fan Characteristics.* As with the centrifugal boiler feed pump, the performance of a fan can be shown graphically by means of characteristic curves.

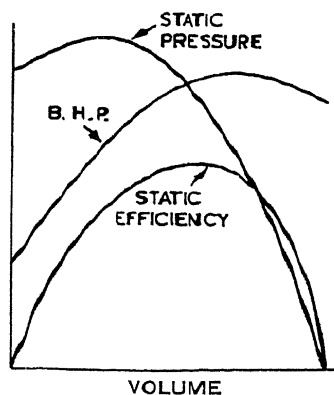


FIG. 124. Characteristic curve of forced draught fan.

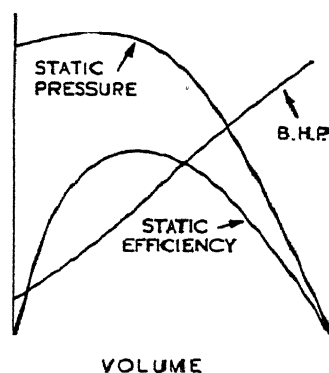


FIG. 125. Characteristic curve of induced draught fan showing diminishing B.H.P. with reduced consumption.

These are usually obtained by plotting static head against the volume discharged at a given speed, and the graphical information is completed by adding the corresponding curves of static efficiency and B.H.P., as indicated in Fig. 124. The characteristics shown are for a forced draught fan, and it will be noted that the pressure curve falls away rapidly after a maximum value. This is a

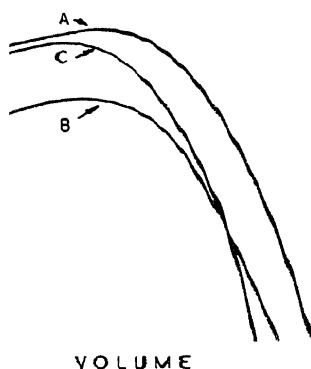


FIG. 126. Fan characteristics with different methods of control.

useful feature for forced draught work, as it means that the resistance must vary considerably before the volume delivered is appreciably affected, and it is also useful when two or more fans have to work in parallel.

A further point is that the B.H.P. curve attains a maximum value and then recedes, and this self-limiting characteristic prevents damage to the driving motor should the resistance suddenly drop to zero value. The difference between the characteristics shown in Fig. 125 and the curves in the preceding

figure should be noted. The pressure-volume curve is much less steep for the radial fan, and the B.H.P. increases as the volume discharged rises, so that provision must be made to prevent over-loading. The efficiency, also, is slightly less than that of the forced draught fan.

*Fan Control.* To operate a fan at maximum economy, some form of control must be introduced to relate the pressure and volume of the air delivered to the actual amount required. Control can be effected by varying the speed of the fan, by closing the outlet damper, or by some form of inlet-vane control. Varying the speed of the fan results in the production of another characteristic curve, B (Fig. 126), in which the pressure corresponding to a given volume is less than in the original characteristic, A, and power is saved. By partially closing the outlet damper while maintaining the original fan speed, the pressure on the discharge side is prevented from falling on account of the increased resistance offered by the damper, and power is needlessly wasted in overcoming the artificially-produced damper resistance.

The use of vane control at the fan inlet, in conjunction with a constant fan speed, allows the fan to work on a number of fan characteristics, as determined by circumstances. Thus, for the conditions depicted in Fig. 126, the application of vane control would cause the fan to operate along the characteristic, C ; for lower output conditions the vanes would be closed still further, and the fan would transfer to another and lower characteristic. The difference in power economy between speed and vane control depends upon the type of driving medium and its efficiency, and upon the fan characteristics.

## CHAPTER XV

## BOILER FEED WATER PURIFICATION

Mineral salts in water—Effects of scale—Water treatment—The lime-soda process—The zeolite process—Demineralisation—Internal treatment and water conditioning—Feed water for high pressure boilers—Corrosion—De-aeration—Caustic embrittlement—Blow-down, its control, technique and disposal—Control of water treatment—Sampling and testing.

ON its journey from cloud to earth, rain becomes saturated with the gases of which the atmosphere is composed. In the vicinity of industrial areas, where there is atmospheric pollution, gases such as ammonia ( $\text{NH}_3$ ), hydrochloric acid ( $\text{HCl}$ ), sulphur dioxide ( $\text{SO}_2$ ), etc., will be taken into solution in addition to carbon dioxide ( $\text{CO}_2$ ) and oxygen ( $\text{O}_2$ ). The latter are, however, the most important, and rain when it reaches the earth may contain as much as 15 cubic centimetres of carbonic acid gas ( $\text{CO}_2$ ) and 30 cubic centimetres of free oxygen per gallon. This dilute solution of carbonic acid dissolves limestone—which is insoluble in water free from  $\text{CO}_2$ —together with other soluble salts, and carries away insoluble matter in suspension as silt.

Some of these substances form solid deposits as the water is heated to boiling point; others are not deposited until a higher temperature is reached or water is removed by evaporation. A water may therefore form deposits in both economiser and boiler and, as might be expected, waters from different sources show wide variations in behaviour when used as boiler feed; some cause corrosion and little scale, others deposit thick scales of varying composition and texture. On the other hand, some waters can form soft deposits in the economiser and hard deposits in the boiler, the composition of the one differing radically from that of the other.

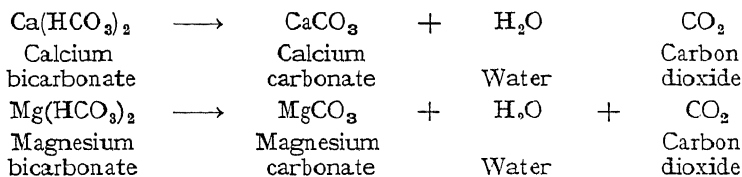
It will be readily understood, then, that no natural water is entirely suitable for use as boiler feed, and some form of treatment is required to counteract the effects of the dissolved salts.

The chief scale-forming substances which are deposited in boilers are compounds of calcium and magnesium (lime and magnesia) and as these substances are also the cause of hardness in water, it follows that hard waters usually form large amounts of scale, although not all the hardness salts take part.

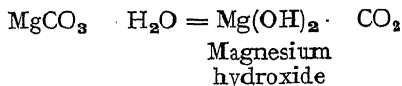
The calcium and magnesium salts in natural waters may be divided into two groups:—

- (1) The bicarbonates, which form what is commonly known as “temporary hardness.”
- (2) The sulphates, chlorides and nitrates, which form what is commonly called “permanent hardness.”

When the water is heated, the bicarbonates are decomposed, giving off carbon dioxide and the resulting calcium and magnesium carbonates are deposited as solids.



Magnesium carbonate is more soluble than calcium carbonate, but it is decomposed under boiler conditions to form magnesium hydroxide, which is less soluble than calcium carbonate.



A water containing temporary hardness will therefore deposit solids whenever the temperature rises to about 70° C. (158° F.), i.e. in economisers, preheaters, etc. as well as in the boiler itself.

The permanent hardness salts do not form deposits when the water is boiled at atmospheric pressure but under boiler conditions they concentrate in the boiler water. They are highly soluble with the exception of calcium sulphate, which crystallises out on the heating surfaces, and this substance is the chief constituent of thick hard boiler scales. Silica is also present in natural waters, though only to a small extent compared with the hardness salts, but it accumulates in the boiler water and thin hard scales of calcium and magnesium silicates can be formed. The magnesium permanent hardness salts decompose in the boiler and give rise to acidic conditions which may cause severe corrosion.

### EFFECTS OF SCALE

It has long been recognised by boiler engineers that natural waters form scale in boilers and in economisers or feed water heaters. The amount and texture of the scales formed vary widely from one supply to another. Some scales are hard and very difficult to remove mechanically; others are soft and easily removed; there are also waters which form practically no scale but deposit sludges which can easily be removed from the boiler by means of a hose. It has become recognised boiler practice to shut down boiler plant periodically for the purpose of removing the solid deposits, and the expenses incurred will of course, depend on the frequency of shutting down and the difficulty of removing the scale.

The first effect of scale formation is therefore a recurrent expenditure in labour and a loss on capital while the plant is shut down for cleaning, together with the attendant disadvantages of having the boiler, and possibly other plant dependent on it, out of commission.

In view of the many statements which have been made regarding the loss of efficiency due to scale formation in a boiler, it seems worth while to try to get a clearer idea of the effects of scale. In a boiler the formation of scale on the heating surfaces reduces their conductivity and the rate of heat transfer to the water so that in scaled multitubular boilers the unused heat in the gases leaving the hottest banks will be absorbed by the more remote banks of tubes, and if these in turn became scaled the region of heat transfer would be removed to the economisers and preheaters. However, tube failure in the hotter regions of the boiler will usually occur long before this process is carried to the limit, particularly in boilers where the greater part of the heat is transferred by radiation.

In practice, therefore, scale is not likely to cause a big rise in stack gas temperature (with a corresponding large decrease in the efficiency of heat utilisation) in multitubular boilers fitted with economisers and air preheaters. On the other hand, in Lancashire or locomotive boilers not fitted with an economiser, scale on the heating surfaces may result in rise in stack gas temperature with loss of efficiency. The effect of scale on fuel consumption is therefore a function of the design of the boiler plant, and in general the simpler the design the greater the effect of the scale. The most important consideration is that the temperature of the metal increases and eventually reaches the point at which mechanical failure occurs (cf. Chapter VIII, pp. 121 and 126).

Many figures have been published for the loss in heat transfer due to varying thicknesses of scale, and a study of these has been made by E. P. Partridge

(*Engineering Research Bulletin*, No. 15, June, 1930: "Formation and Properties of Boiler Scale," Department of Engineering Research, University of Michigan). The following are quotations from the conclusions in this investigation:—

"Boiler scales have low heat conductivities; the maximum value of the heat conductivity coefficient is approximately 14.0 B.Th.U./sq. ft./hr./°F./in. Dense, compact scales, such as are typically formed by calcium sulphate, show an average value of approximately 15, while very porous scales may have coefficients as low as 0.6. The extremely low conductivity of porous scales is due to the fact that during boiler operation the interstices are filled with steam, forming what is, in effect, a vapour film mechanically held on the boiler heating surface. The high resistance of vapour films to heat flow is well known.

"While boiler scale is a material with low heat conductivity, its effect upon heat utilisation is slight. Older figures for heat loss due to scale range up to 15 per cent. loss for  $\frac{1}{16}$  inch of scale, but it is probable that the actual loss is not more than 2 per cent. The latter figure is supported not only by theoretical calculations, but also by the result of recent experimental investigations.

"While the loss in boiler efficiency due to scale is small, the fact that there is any loss at all would still justify the application of scale prevention methods. Contemporary boiler operation demands the elimination of all losses up to the point of maximum economy of operation. The prevention of scale by adequate chemical control would be sound economics even if boiler efficiency were the sole consideration. Actually, however, there is another factor which is growing in importance as boiler pressures and rates of driving steadily increase. This is the failure of tube surfaces due to overheating as a result of scale formation."

An illustration of this was given in Chapter VIII when dealing with thermal conductivity. It was there calculated that when the safe working maximum temperature for boiler tubes is 900° F., a coating of scale of only  $\frac{1}{8}$  inch thickness would be sufficient to cause this limiting temperature to be reached. The scale in this example had a thermal conductivity of 18 B.Th.U./sq. ft./hr./°F./in. thickness. Obviously if the scale had been of the porous variety, a very much lesser thickness than this would have led to tube failure.

The losses in money and time occasioned are sufficient themselves to warrant careful examination of any methods of preventing scale which are simple and relatively inexpensive, and for many years increasing attention has been paid to the conditioning of boiler feed water by the addition of suitable chemicals.

## SCALE PREVENTION

It is essential to analyse any scales which are formed in a boiler as a knowledge of their composition is a valuable aid in devising water treatment for their prevention.

Chemical analysis has shown that the chief constituents of hard scales are calcium sulphate and/or calcium and magnesium silicates, and that the soft loose scales or sludges are composed of calcium carbonate and magnesium hydroxide. Broadly speaking, permanent hardness forms hard adherent scales; temporary hardness forms soft scales or sludges. It follows that the best method of preventing deposits is to subject the water to a preliminary treatment which will effect the removal of scale-forming salts and of substances that may cause corrosion. Any physical or chemical means whereby this can be done cheaply and efficiently can be used.

## METHODS OF WATER TREATMENT

There are three methods of treating water for boiler feed and these can be modified or combined to suit the type of water or boiler operating conditions.

(1) Precipitation processes, in which chemicals are added to precipitate calcium and magnesium as compounds of low solubility. The lime-soda

process is typical of this class, but other precipitating agents such as caustic soda and sodium phosphate can be used where the composition of the water permits and having regard to its ultimate use in boilers.

(2) Processes in which the hardness salts are removed without visible precipitation. The zeolite or base-exchange process may be taken as an example of this class.

(3) Both the above processes leave sodium salts in solution, but these do not cause hardness or scale. All salts can be removed by distillation or the process described below under the head "Demineralisation."

A brief and very general description of these processes is given below, but it must be made clear that the choice of process depends on the composition of the water supply and other factors such as cost, boiler working pressure, etc., and the decision on such points is best left to an expert as indicated at the end of this chapter.

### LIME-SODA PROCESS

In this process hydrated lime and sodium carbonate are added to the water, whereby the calcium and magnesium are precipitated and removed by sedimentation and filtration. It is known as the lime-soda, or lime-soda-sodium aluminate process.

The process can be carried out in the cold but better results are obtained at elevated temperatures, preferably above 70° C. (158° F.). Plants should incorporate automatic apportioning gear, to ensure accurate addition of reagent to the correct volume of water, ample retention time for settling the precipitates—about three hours cold and two hours hot—and filters to remove the final traces of suspended solids. By the hot treatment the hardness can usually be reduced to negligible proportions, i.e. of the order of 1.0 part  $\text{CaCO}_3$  per 100,000, but in the cold such a low figure is not reached except under very favourable circumstances.

When the volume of water to be treated is small, the treatment may be carried out in a tank, or in two tanks used alternately, so that the clear soft water from one may be fed to the boiler while the precipitates are settling in the other. The capacity of the tanks should be sufficient to allow the settling times suggested above, and provision should be made for withdrawing the settled precipitates from the bottom of the tanks periodically. Some precipitate should, however, always be left as a nucleus on which the precipitates in the next softening can form.

The reagents are usually mixed with water and added as a "milk," and it is most important that the quantities shall be strictly proportioned to the hardness and volume of water treated, a small excess of reagent being added to ensure maximum hardness removal. After the addition the contents of the tank should be stirred for a few minutes to ensure thorough mixing before settling.

Whilst the use of aluminate is optional when the water contains little magnesium, when the water contains appreciable quantities of magnesia, sodium aluminate should always be used as an additional reagent. The magnesium hydroxide precipitate normally takes a considerable time to separate and settle, since it tends to remain in the colloidal state. Sodium aluminate hastens the softening reaction by coagulating the magnesium hydroxide into a large floc which settles rapidly, and thus expedites the removal of magnesia. The floc also entrains particles of calcium carbonate or other matter in suspension as it settles. In this way a stable softened water of low hardness can be obtained in a shorter time than with lime and sodium carbonate alone. The amount of sodium aluminate required is small, about 1.0 part per 100,000 in hot and 2.0 parts in cold softening. A reduction in the silica content may

also be obtained when aluminate is employed as an auxiliary reagent and the necessary slight excess of softening reagents is used.

For very soft waters the lime-soda process of softening is not suitable. Phosphates can be substituted for lime-soda in the treatment of these waters, and this process is widely used on the Continent, particularly for high pressure boiler feed. It is carried out at an elevated temperature.

### THE ZEOLITE PROCESS

In this process the water is passed through a bed of mineral which contains sodium in combination and has the property of exchanging its sodium for the calcium and magnesium in the water. The effluent therefore contains only sodium salts and is soft. The calcium and magnesium are retained by the mineral and in due course its capacity for the exchange becomes exhausted. The flow of water is then stopped and the mineral reconverted to its original form by treatment with a relatively strong solution of common salt, the calcium and magnesium passing into the brine and being rejected with it. The cycle of operations is therefore (1) softening, (2) regeneration with brine, (3) draining and washing to remove brine, after which the cycle (1)→(2)→(3) is repeated. Artificial materials having the same properties as the mineral formerly used have now largely replaced natural zeolite.

The process is intermittent in character, but of course can be made to give a continuous flow of softened water by using two or more units alternately. It is carried out in the cold, there is no sludge, and the hardness is reduced to about 0.5 part  $\text{CaCO}_3$  per 100,000, or less under favourable conditions. Since the calcium salts are replaced by sodium salts it follows that the temporary hardness is converted into sodium bicarbonate which decomposes and gives off  $\text{CO}_2$  when the water is heated. Such a water may cause corrosion in the boiler and auxiliary equipment, and the process is not recommended for waters of high temporary hardness which are to be used as boiler feed.

The water supply must be clean, as the bed of exchange material is adversely affected by the presence of suspended matter or soluble substances which may deposit from solution in the bed, such as iron and alumina. When installing a softening plant of this type, therefore, careful consideration should be given to the quantities of these materials which may be present in the raw water, particularly in waters which are subject to variations in composition. Preliminary coagulation, sedimentation and/or filtration may be required to protect the zeolite bed.

### DEMINERALISATION PROCESS

This has been made possible by two further discoveries, firstly that some artificial base-exchange materials, prepared from coal and other highly carbonaceous substances will convert all calcium, magnesium and sodium salts into the corresponding acids when regenerated with acid instead of common salt. Secondly, that an entirely different group of substances, of the artificial resin type, have the power of absorbing dilute acids. These substances are regenerated with alkalies such as caustic soda and sodium carbonate.

Thus, by passing a water first through a bed of hydrogen exchange material and then through the acid removing material all salts are removed. In practice it is found that the treated water contains silica, corresponding to that present in the raw water, together with carbon dioxide, oxygen and dissolved salts to the extent of only a very few parts per 100,000.

With the exception of its silica content, water prepared in this way is comparable with that obtained from evaporators. The silica content is not likely to be objectionable in the boiler, provided that conditioning chemicals are added to prevent the formation of silicate scales.



## DISTILLATION

This method yields a very pure water, but it is fairly expensive and is used only when the volume of make-up water is small or the boiler pressure is high. The water derived from steam traps when uncontaminated with oil, etc., is virtually distilled water.

## CONDITIONING TREATMENT

It must be emphasised that in addition to preliminary treatment it is usually necessary to apply chemicals to the final feed water in order to provide the right balance of chemicals in the boiler water to prevent scaling, corrosion, and caustic embrittlement (see later paragraph), or to modify the form of any precipitate in the boiler so that it can easily be removed in the blow-down. Such chemical adjustments are known as conditioning treatment, and their application will depend on the composition of the final feed water and the boiler operating conditions.

## INTERNAL TREATMENT ; WATER SOFTENING IN THE BOILER

Circumstances sometimes arise in which the installation of a softening plant is impracticable. An alternative treatment is available in the form of the addition of chemicals direct to the water entering the boiler, by which means the whole of the hardness is precipitated in the boiler as a sludge instead of a scale.

This method has been applied with very satisfactory results to waters up to 25 parts  $\text{CaCO}_3$  per 100,000 in boilers working at moderate pressures and rating. In general, the treatment is recommended for water-tube boilers only when the hardness of the feed water is low, but in boilers of the Lancashire type water of higher hardness may be treated by this method.

The basis of this treatment is the addition of sodium carbonate or phosphate to the water as it enters the boiler, in an amount sufficient to precipitate the permanent hardness. The temporary hardness is, of course, precipitated by heat. The precipitates are in a finely divided form and may tend to agglomerate and bake on to the heating surfaces. Where they collect in a mud drum it has been observed that they pack, and only a small amount of mud in the immediate vicinity of the blow-down cock passes out when this is opened.

An improvement in these conditions can be effected by the addition, with the sodium carbonate, of tannins or mixtures of sodium aluminate and tannins, by means of which the precipitate appears to be coagulated into a free flowing form which is maintained in suspension by the circulation of the boiler water and can be easily removed by intermittent or continuous blow-down. For waters of low hardness sodium phosphate is preferred as a precipitating agent, and the coagulation of the precipitate is effected by the use of tannins mixed with the phosphate. Similar principles may be used to prevent precipitation in the economisers.

## HIGH PRESSURE BOILERS

The feed water for modern high pressure boiler installations is usually a mixture of condensate and distilled water as make-up. Most waters will form scale in the evaporator and it is advisable to consider a preliminary softening treatment to maintain the evaporator heating surfaces in a clean condition.

The distillate should, of course, be free from salts, but in practice small amounts of carry-over or priming often occur, and contamination of the feed water can also arise from condenser leakage. Conditioning chemicals will be required to counteract the contamination, but should not be regarded as a substitute for mechanical attention to condensers and mechanical repairs

generally. With phosphate conditioning agents it is an advantage to maintain a little free phosphate in the boiler water. These points are highly important.

### CORROSION

All boiler feed waters should be alkaline in order to minimise corrosion, and where the feed consists of distillate and condensate it will be necessary to add a little caustic soda to make the feed water alkaline to phenol-phthalein. The small quantity of caustic soda normally required for this purpose should be added continuously. It should be emphasised, however, that alkalinity alone cannot be relied upon to give adequate protection against corrosion in boilers working at pressures of the order of 250–300 lb. per square inch or above, and the oxygen and carbon dioxide content of the feed water must be maintained at a low value. The suggested limits for the dissolved oxygen content of the feed water are :—

Boilers working at 300–450 lb. per square inch, not more than 0.02 cubic centimetre per litre. At higher pressures every effort should be made to reduce this below 0.01 cubic centimetre per litre.

### DE-AERATION

Mechanical de-aeration is generally effected either by passing the feed water through special equipment installed for this purpose, or by de-aeration in the turbine condenser to which the distillate from the evaporator is introduced and mixes with the condensate.

In practice the residual oxygen content may exceed the recommended figure, e.g. when the de-aerator is working at peak loads, or from other causes of a temporary nature, and the additional oxygen then present in the feed water may be removed by chemical de-aeration with sodium sulphite. The sodium sulphite should be added continuously in the form of an aqueous solution at a point beyond which atmospheric contamination cannot occur. When caustic soda is added to the feed water the two reagents can be fed together and it is advisable to have the chemical solution tank covered with a close-fitting lid to minimise oxidation by contact with the atmosphere. The quantity of sulphite required is based on the oxygen content of the feed water with an excess sufficient to give a reserve of 3–5 parts  $\text{Na}_2\text{SO}_3$  per 100,000 in the boiler water.

### CAUSTIC EMBRITTLEMENT

The cracking of boiler metal as a result of attack by caustic soda has been the subject of extensive study on the laboratory and plant scale. These cracks can be identified under the microscope because they are intercrystalline and show no marked deformation of the grain, whereas those due to corrosion or fatigue failures are transcrystalline and are associated with grain deformation.

The investigations have shown that at least three factors must coincide in order to produce the characteristic intercrystalline attack.

- (1) Unequal stress distribution in the metal, with high local stresses such as might occur in riveted joints due to misalignment of rivet holes, deformation of rivets, etc.
- (2) The occurrence of crevices in the neighbourhood of the highly stressed metal, e.g. overlap joints, etc.
- (3) The leakage of boiler water containing caustic soda into the crevices and its concentration therein in contact with the stressed metal. It has been demonstrated that such concentration can occur under boiler conditions especially when leakage to the outside occurs. Small leaks which are invisible but can be detected by condensation on a cool surface appear to be able to provide the necessary concentration conditions.

The attack usually takes the form of irregular crazy cracking on the dry side

of the plate starting from and running between rivet holes. Although the rest of the metal is quite sound the development of these cracks obviously leads to serious weakness which can result in sudden failure at the joint.

It is clear that both mechanical and chemical factors are involved and that the mechanical factors are particularly difficult to detect in a working boiler. Since leakages provide a means of concentration of the boiler water, however, it follows that the detection of small leaks is an important part of boiler maintenance and that any boiler with a history of leakage is suspect and should be carefully examined by an expert metallurgist. It is recommended that joints should be caulked internally and that welded drums should be stress relieved. There appears to be little evidence of embrittlement in operating boilers with forged drums.

Certain substances, such as lignins, tannins, sodium sulphate, etc., have been shown to have a retarding effect and up to the present it has been common practice in the United Kingdom to maintain definite concentrations of sodium sulphate in the boiler water as a safeguard. There are wide differences of opinion regarding the effectiveness of this chemical but at the time of writing no alternative material has been definitely established by large-scale experience as superior. It seems reasonable therefore to continue to use sodium sulphate for the present, and sufficient of this substance should be added to maintain in the boiler water at all times a weight ratio sodium sulphate/caustic soda ( $\text{Na}_2\text{SO}_4/\text{NaOH}$ ) greater than 2.5, when riveted drums or welded drums which have not been stress relieved are used. Many waters contain sufficient sodium sulphate for this purpose, but other waters require a special addition of this chemical. It must be emphasised that no chemical additions to the boiler water can heal cracks which have once started because even if embrittlement factors have been removed from the water, the cracks act as stress raisers and failure may occur as a result of corrosion fatigue.

It is clear that the present position regarding this subject is not very satisfactory from the point of view of the engineer in charge of boilers, and it should be pointed out that boiler failures as a result of caustic embrittlement are not common, but that this does not justify neglect of the precautions which appear to be effective in the present state of knowledge. Further developments both in laboratory research and practical operating experience will no doubt be published from time to time and should be carefully studied by those responsible for boiler plant.

## BLOW-DOWN

A boiler is blown down to prevent the unrestricted accumulation of suspended and dissolved solids which would lead to contamination of steam with boiler water, commonly called priming, foaming, or carry-over, and ultimately to the crystallisation of the very soluble sodium salts. It is an essential feature of all boiler operation whatever treatment is applied to the feed water.

## AMOUNT OF BLOW-DOWN

The amount of blow-down should be controlled by the amount of total dissolved solids (hereafter referred to as T.D.S.) in the boiler water, which should be consistently maintained slightly below the maximum permissible amount. This maximum figure requires determination under normal operating conditions for each boiler plant and make of boiler. It is affected by such factors as the amount of suspended matter in the boiler water, the variability of the boiler load, and the level of the water in the drum. The highest figures will be obtained when :—

- (1) The boiler load is steady.

- (2) The boiler water is clean, i.e. free from all but small amounts of finely divided suspended matter.
- (3) The water level is just above the centre line of the drum and not higher than, say, two-thirds the diameter of the drum.

The determination of the maximum permissible T.D.S. depends on observations of steam purity. When large quantities of boiler water are present in the steam the superheat temperature falls suddenly. This phenomenon is often called priming. There appears to be another condition, however, in which small amounts of boiler water pass off with the steam more or less continuously, and this occurrence is often known by the term "carry-over." It is doubtful whether the carry-over of small amounts of boiler water will appreciably affect the superheat temperature, and the method of detection usually employed is the measurement of the electrical conductivity of steam samples, i.e. the steam purity.

For this purpose a sampling nozzle (bore about  $\frac{1}{16}$  inch for 175 lb. pressure and 6-inch steam pipe) is fitted into the steam line pointing up-stream and on the dead centre of the pipe; sharp bends in the pipe are to be avoided. The nozzle is connected to a cooling coil of sufficient capacity to reduce the temperature of the sample to atmospheric. The flow of the condensate must be adjusted so that the linear velocity of steam in the sampling nozzle is the same as that in the steam line from which the sample is taken.

When the boiler is working on normal load the conductivity of several steam samples should be measured and then, with the blow-down valve shut, 24-hour average samples of steam condensate and boiler blow-down can be taken and the conductivity of the steam sample plotted against the concentration of dissolved solids in the blow-down. That concentration at which a sharp increase in conductivity of the steam sample occurs may be regarded as the maximum permissible total dissolved solids content in the boiler water for the particular type of boiler and operating conditions. The effect on the maximum permissible T.D.S. of the other factors already mentioned may be observed by a similar set of tests. In practice it is found that certain maximum figures are fairly generally applicable and these are given later in Table 68. The actual T.D.S. must be maintained below the maximum (cf. p. 245).

If the maximum permissible T.D.S. in the boiler water is A parts per 100,000 and the T.D.S. in the feed water entering the boiler is B parts per 100,000, then the theoretical blow-down will be  $\frac{100B}{A}$  per cent. of the total evaporation.

## MEASUREMENT OF T.D.S.

There are four methods in use :—

- (1) By hydrometer. These can be obtained from suppliers of laboratory apparatus, but they should be checked against actual determinations of T.D.S. by evaporation to dryness and weighing. When ordering, the purpose for which the hydrometer is required should be stated.
- (2) By the conductivity of the blow-down water. This figure can be used alone after calibration against actual determinations. The method permits of a continuous record being kept when continuous blow-down is installed.
- (3) By determination of chloride in both feed and boiler water. Chlorides are very soluble and are not eliminated as scale and, therefore, this chemical test is a fair indication of concentration when the composition of the feed water is fairly constant, but it is not as reliable or as easy to carry out as methods 1 and 2.
- (4) Evaporation to dryness and weighing. This is a standard laboratory method rather tedious and unnecessary for routine plant control.

## BLOW-DOWN TECHNIQUE

Continuous blow-down is the most satisfactory method of removing water from the boiler because it can easily be adjusted to maintain the T.D.S. in the boiler water consistently at the desired level, and takes the matter out of the hands of the operator. The method preferred is to fit a short horizontal length of pipe below the low-water level in the steam drum and at the end opposite to the feed entry. In cylindrical boilers an open-ended vertical pipe may be used and the open end should be more than 2 inches above the top level of the flue tubes so that the boiler cannot blow itself dry. A suitable size of orifice, or a special fine control valve, is fitted in the blow-down line outside the boiler so that the amount can be controlled. If the boiler water contains appreciable amounts of suspended matter the ordinary blow-down valve should be opened momentarily once per day to remove accumulated solids. Continuous blow-down is not recommended when the volume of water to be removed is less than 25 gallons per hour.

When the blow-down is intermittent and effected by means of the usual valve fitted at the lowest point of the boiler, this valve should be used as frequently as is practicable, taking into account the total volume of water to be removed. If possible, the valve should be opened once per shift and usually not less than once per day. In this way the composition of the boiler water will be kept as constant as possible. To ensure the maximum removal of accumulated sludge, the valve should be opened momentarily several times with a short pause in between so that the disturbance in the neighbourhood of the blow-down outlet can die down and fresh sludge can move into position ready to be flushed out. When starting up, boilers should not be blown down until the T.D.S. reaches the proper value, and it will be found advisable to have a clear set of instructions issued to each boiler house covering all conditions of operation.

## DISPOSAL OF BLOW-DOWN

There are four methods of disposal in common use, viz. :—

- (1) To waste : if this method be adopted the most rigid control is necessary to economise in fuel.
- (2) Through a heat exchanger, in which the feed water is heated, or alternatively through a flash vessel from which the steam is passed direct into the feed water and the heat in the residue is recovered by a heat exchanger. A further economy of fuel can be secured by reducing this residue to a minimum.
- (3) Through a flash vessel and heat exchanger as before, but part of the residue from the flash vessel is returned to the softening plant where its heat and chemical content are used.
- (4) There is a fourth method which can be applied to boilers in which the feed water is condensate and evaporated make-up, namely, by returning a portion of the blow-down direct to the feed so that the chemicals in it are used for conditioning purposes.

While the importance of heat recovery by exchange is obvious, it is not always practicable with existing plant layout, particularly at the present time when economy of steel is also required. The installation of heat recovery plant may have to be decided on the basis of steel economy versus potential fuel economy, and careful consideration will be required in each case, but there is no doubt that much can be done to limit heat losses when the blow-down is sent to waste, and the data collected with this object in view will also be necessary for a consideration of heat economy by the use of exchangers.

## CONTROL OF WATER TREATMENT

It should be emphasised that satisfactory softening and conditioning can be attained only by careful control, and routine testing is therefore essential. Simple and rapid methods of analysis have been developed for control purposes. For testing purposes a clean and well-lighted place should be provided since many of the erratic results in softening plants and with boiler feed water conditioning may be traced to errors in testing due to dirty testing apparatus, bad light, etc.

Except where otherwise stated, all tests should be carried out on the sample after filtration through filter paper.

A summary of the necessary tests is given below and for convenience these have been grouped under two main headings:—

- (1) Water softening.
- (2) The control of blow-down and boiler feed water conditioning.
- (1) The determinations necessary for the control of a lime-soda water softener are as follows:—

### (a) *Raw Water*

Temporary hardness.

Free  $\text{CO}_2$  (not essential, as an extra quantity of lime can be added to combine with  $\text{CO}_2$ ).

Total hardness.

Hardness due to lime salts.

Hardness due to magnesia salts (found by difference).

### (b) *Softened Water*

Total alkalinity to methyl orange indicator.

Caustic alkalinity.

Residual hardness due to lime.

Residual hardness due to magnesia.

Hardness in suspension (sometimes necessary to check the efficiency of filtration in lime-soda plants).

## Raw Water

These determinations should be carried out once a day and the quantities of reagents added should be calculated from the analysis of the raw water. If it is found that the composition of the raw water does not vary appreciably from day to day, then less frequent analyses may suffice.

## Softened Water

These determinations are required for the control of the softening plant and should be carried out twice per day unless the raw water does not vary appreciably in composition and more often if the water is variable.

The sample of softened water should be taken from the softening plant outflow pipe some four hours after any change in the addition of reagents, in order to allow sufficient time for the change to become effective.

Zeolite softening plants are usually operated on a volume or time basis. The plant consists of several units each containing sufficient material to soften a definite volume of water of given composition. Provided that this composition remains constant it is then only necessary to carry out determinations of alkalinity and total hardness on the raw water once per day to establish that the composition is up to specification, or, if the composition varies, to alter the

volume passing through the unit before regeneration. The softened water should be tested in the same way several times immediately before it is expected that the unit will become exhausted, and as soon as the residual hardness in the softened water rises to about 1.0 part  $\text{CaCO}_3$  per 100,000 the unit should be regenerated.

(2) The determinations necessary for the control of boiler feed water conditioning and boiler blow-down are as follows:—

(a) *Feed Water*

(Once per day or more often if required. Feed water is often the softened water already tested.)

Caustic alkalinity.

Chloride.

Sulphate.

Oxygen (for high pressure boilers a recorder is advised).

Hardness.

(b) *Boiler Blow-Down*

Total alkalinity.

Caustic alkalinity.

Chloride.

Sulphate.

Phosphate (where phosphate conditioning is used).

Hardness. (This is a detection test and not an estimation.) The detection of hardness indicates the possibility of scale formation and extra chemical treatment should be applied at once.

Sulphite (where sulphite is used for chemical de-aeration).

Total dissolved solids.

From these determinations it will be possible to ascertain without difficulty whether the composition of a boiler water satisfies the conditions laid down for the prevention of corrosion, caustic embrittlement and scale formation. The suggested tests may at first sight seem somewhat formidable, but for simple types of boiler a simpler scheme of tests can be worked out; similarly for large boiler plants the number of tests can be reduced to a minimum by careful planning.

It will be appreciated that the characteristics of the feed and boiler water will require to be maintained at values which experience has shown to be necessary for the working pressure and type of each particular boiler installation. In general terms the specifications for the feed, softened make-up and boiler waters in low, medium and high pressure plants are as given in Table 68.

TABLE 68. SPECIFICATIONS FOR BOILER FEED WATER AND WATER IN BOILERS

	Softened water	Feed water
Total alkalinity ..	Approx. 8 pts. $\text{CaCO}_3$ /100,000	Alkaline to phenol-phthalein at all times.
Caustic alkalinity..	„ 3 pts. $\text{CaCO}_3$ /100,000	
Hardness .. ..	1 part $\text{CaCO}_3$ /100,000 as near as possible.	As low as possible.
Oxygen .. ..		Medium pressure not exceeding 0.02 c.c. per litre. High pressures not exceeding 0.01 c.c. per litre.

	Boiler waters		
	Low pressure up to 250 lb./sq. in.	Medium pressure 250–500 lb./sq. in.	High pressure above 500 lb./sq. in.
Total alkalinity ..	15–20% of T.D.S.	15–20% of T.D.S.	15–20% of T.D.S.
Hardness (Wanklyn foam test) .. ..	Zero.	Zero.	Zero.
Sulphate .. ..	Not below $2\frac{1}{2}$ times caustic alkalinity.	Not below $2\frac{1}{2}$ times caustic alkalinity.	Not below $2\frac{1}{2}$ times caustic alkalinity.
Phosphate .. ..	$8\frac{1}{2}$ –17 parts per 100,000 as $\text{Na}_3\text{PO}_4$ .	$8\frac{1}{2}$ –17 parts per 100,000 as $\text{Na}_3\text{PO}_4$ .	$8\frac{1}{2}$ –17 parts per 100,000 as $\text{Na}_3\text{PO}_4$ .
Total dissolved solids (T.D.S.) for blow-down control.	Up to 700 parts/ 100,000.	500–200 parts per 100,000.	Not exceeding 100 parts/100,000.

(N.B.—For Economic boilers maximum T.D.S. recommended is 400 parts/100,000.)

### ROUTINE TESTING OF WATER SAMPLES RECOMMENDED PROCEDURE FOR SAMPLING

It will be realised that the method adopted in collecting samples of water for testing will have an important bearing on the control of water softening and of feed and boiler conditioning since the adjustment of the treatment is based on the results so obtained. No fixed system of sampling can be laid down to cover all boiler plants but the following notes, which are intended as a general guide, may be of use in developing a proper scheme of sampling.

The samples should, if possible, be obtained from permanent sampling points and these should be so arranged that there is no doubt about the samples being representative.

### RAW WATER

More often than not raw water supply is subject to fluctuations in composition, these variations being dependent on a number of factors. Daily average samples are therefore only useful for control purposes when the variations are small. Variable waters must be sampled as frequently as required for the control of plant. It is important to take samples at all times of the year so as to build up a history of the water composition.

### SOFTENED WATER

Spot samples are required for plant control and daily average samples for checking and record purposes. The main precaution is that the sample should be taken at a convenient point on the softened water outlet line as near to the softening plant as possible.

### FEED WATER

Where the feed water consists of condensate and make-up water mixed in a feed tank or hot well, it is inadvisable to assume that a sample taken from the feed tank is entirely representative. A sample taken from the boiler feed pump is therefore to be preferred.

### CONDENSATE

Sampling of condensate presents rather more difficulty as although in general there are numerous points at which samples can be obtained, it is necessary to sample at a point where the various condensate returns mix together before re-entering the feed systems. If contamination occurs each source of condensate should be sampled.



## BOILER WATER

Boiler water samples are usually taken from the gauge glass. This is done by shutting the steam cock on the top of the gauge glass, opening the gauge glass drain, and allowing the water to run to waste until the connecting pipe between the bottom of the gauge glass and the boiler is flushed through before the sample is taken. This is very important, as the continual condensation of steam in the gauge glass from the upper steam connection results in the water lying in the connecting pipe to the boiler being mixed with condensate.

In boilers fitted with continuous blow-down, the discharge to waste pipe provides a convenient sampling point. A sample taken from either point has the disadvantage of being subject to steam "flash-off" unless suitable precautions are adopted. This "flash-off" can be successfully obviated if the sample is passed through a cooling coil, and this method of sampling is advised.

In boilers receiving individual treatment it is advisable to test each boiler separately. Where there are a number of boilers utilising the same feed water supply it is suggested that they should be divided into groups of four, one boiler from each group being tested in rotation each day. This ensures that each boiler water is tested at least once every four days.

It is preferable to collect the boiler water samples in a copper cylinder or a welded sheet-iron bottle fitted with a close-fitting lid or stopper. The samples should be stoppered immediately, cooled as quickly as possible, and then transferred to a bottle and immediately stoppered. By this means the samples can be kept and stored without undergoing change until it is convenient to carry out the tests.

## WORKS CONTROL

From this discussion it may appear that the treatment of water and its control is a somewhat complicated matter, but this is due to the attempt to cover the essential features of the process as applied to a variety of boiler plants. After a study of any individual boiler plant the picture which is revealed is usually fairly simple. Boiler plants have an individuality in design, layout, operation and staff and a study of these points in each plant is therefore essential so that a scheme of water treatment can be devised to fit the existing conditions. Long experience has shown that failure to devise such a scheme is rare provided that there is first a proper understanding of the special conditions at the plant concerned and that all changes in conditions which can be foreseen are taken into account.

A study of water conditions at a boiler plant should be carried out in three stages.

### (1) PREPARATION

An appreciation should be made of the nature of the water supply, e.g. source, composition, variability, contaminations, etc., and of the boiler plant, e.g. size, type of boiler, working pressure, layout, etc.

### (2) DESIGN

A scheme of water treatment and control testing should be devised, based on the appreciation of the factors in (1).

### (3) OPERATION

The whole scheme should be discussed with the staff concerned and modified where possible to meet special objections or requirements. At least two members of the staff should be instructed in the basic principles of the treatment and the tests required, and when the scheme is put into operation various check tests will be necessary before it can be established on a routine basis. Minor modifications in treatment may have to be made in the light of operational experience.

It is highly desirable that some member of the staff should be made responsible for the maintenance of the water conditions laid down in the scheme and for the testing required as well as for the study of the principles involved. He should act in an advisory capacity to the manager of the boilers and should work in close co-operation with him. An arrangement of this kind is particularly necessary when the boiler plant is large or a number of boiler plants comes under one control.

It is clear that the development of such a scheme requires a thorough knowledge of the principles of water treatment at all stages of its progress from source to boiler, coupled with judgment and experience.

### NEED FOR EXPERT ADVICE

The account here given of boiler feed water treatment is necessarily sketchy and incomplete. Reference is made particularly to the purification of feed water for low and moderate pressures ; for high pressure boilers distilled water free from dissolved gases must be used.

Sufficient will have been said to indicate that the treatment of boiler feed water is primarily a matter for a chemist and that the successful operation of a softening plant depends on chemical control, just as the installation of the right type of plant to meet local circumstances requires expert advice.

This advice is best obtained by employing a chemist. When that is impossible, a consultant may be able to help materially by paying periodic visits and making the necessary tests or teaching the boiler staff to carry out simple control tests. Many small boiler plants are controlled in this way after preliminary consultation with an expert. If no advice can be obtained, the technical staff of the Ministry of Fuel and Power are always ready to assist enquirers

## CHAPTER XVI

**BOILER EFFICIENCY AND TESTING**

The methods of testing steam boilers to secure essential information regarding operating results—Examples of operating results for efficient and inefficient plants—The more detailed examination of boiler plants.

**T**HE object of those in charge of boiler plant should be to realise the highest possible efficiency with the plant and fuel at their disposal. It follows that to know whether this object has been achieved, it is necessary to know what is possible with the particular plant. There is no other way of determining what is possible than by testing the performance of the plant.

If a proper standard of performance is established for a particular boiler, any falling off in performance can be detected and the cause traced.

The procedure in boiler testing will depend upon the extent of the information that is required and also, incidentally, upon the size and skill of the staff available for the work and the extent to which they are provided with instruments. There may well be a difference in procedure between routine testing undertaken to keep the plant in good order and a more comprehensive test undertaken at much longer intervals to ascertain a complete heat balance. Here the simple procedure is dealt with on the basis of British Standard Code 845. This code is not intended for use in the comprehensive tests, such as would be needed for large boilers, but it is accepted here as being a useful guide for routine testing for the majority of works boiler installations. Comprehensive tests can be performed on the basis of Table 73, or by using the code drawn up by the Institution of Civil Engineers.

The object here is to indicate the methods which should be adopted and the data which it is desirable to secure when carrying out a simple efficiency test at minimum cost on steam-raising plants using solid fuel, to obtain a satisfactory measure of performance. A warning must be given to the effect that if the load on the boiler fluctuates, the period over which the test is run must be sufficient to embrace a complete cycle of such fluctuations. In B.S. 845 the readings taken and the calculations made are reduced to the minimum. As will be seen from the test results given later in this chapter, this method determines the overall thermal efficiency, and the losses in flue gases and ashes. All other losses are taken as a difference figure. The results, moreover, are referred to the net C.V. of the fuel. The Institution of Civil Engineers code is based on gross C.V., and the gross C.V. is generally to be preferred.

Some specialised knowledge is required to ensure that reliable results are obtained by testing. Such knowledge can only be acquired by practice, but practice is often difficult to obtain, and therefore it is advisable to secure assistance for at least the first test from an engineer who has had practical experience in boiler testing, and who will advise as to details of methods, instruments, and how to avoid inaccuracies of measurement. Such assistance is readily available and can be obtained upon application to the Ministry of Fuel and Power.

Before making a test to set up a standard of performance, it is essential that the boiler and its auxiliaries shall be in such condition that all avoidable losses are eliminated. This means that all heating surfaces, both internal and external, must be clean; that no leaks of water or steam occur; that no air leaks occur through the setting of the boiler or the flues; that the grate is in good condition, and that the air supply is effective.

The fuel used for a performance test should be representative of that which will normally be used.

The thermal efficiency of a boiler may be determined in two ways by :—

- (a) Direct measurement, and
- (b) Indirect measurement.

The direct method should give better and more positive results and it necessitates the measurement of the fuel burned and of the water evaporated. While this method is recommended it is recognised that in some works it may not be possible to make suitable arrangements to determine the weight of fuel and water. In such cases it is necessary to make use of the indirect method, which consists of ascertaining the various losses, and after expressing these as percentages, deducting their sum from 100; the resultant then gives the thermal efficiency of the boiler plant.

An example (Table 69) will be given set out in the manner recommended by B.S. 845.

TABLE 69. EXAMPLE 1

B.S. Item	7.	Boiler water tube heating surface ..	sq. ft.	Not required for this calculation.
		Superheater integral heating surface	sq. ft.	Ditto.
		Economiser heating surface .. ..	sq. ft.	Ditto.
10.		Air heater (not installed).		
11.		Method of firing		Travelling grate.
12.		Grate area. .. ..	sq. ft.	140
13.		Draught plant (chimney draught only)		Not required for this calculation.
14.		Duration of test .. ..	hrs.	8
<i>A. Fuel</i>				
15.		Description .. ..		Northumberland slack.
16.		Proximate analysis as weighed :—		
		Moisture .. ..	per cent.	12.4
		Volatile matter .. ..	.. ..	28.5
		Fixed carbon .. ..	.. ..	47.7
		Ash .. ..	.. ..	11.4
			.. ..	100.0
17.		Gross or higher calorific value as weighed ..	B.Th.U./lb.	11,000
18.		Net or lower calorific value as weighed ..	B.Th.U./lb.	10,478
19.		Fuel fired per hour .. ..	lb.	2,754
20.		Fuel fired per hour per sq. ft. of grate area	lb.	19.7
<i>B. Ashes and Clinkers</i>				
21.		Collected per hour .. ..	lb.	342
22.		Percentage of fuel fired .. ..		12.4
23.		Percentage of combustible .. ..		8
<i>C. Water</i>				
24.		Temperature entering economiser .. ..	° F.	142
25.		Temperature entering boiler .. ..		Not required as economiser fitted.
26.		Evaporation per hour .. ..	lb.	20,152
27.		Evaporation per lb. of fuel as weighed	lb.	7,318
<i>D. Steam</i>				
28.		Gauge pressure .. ..	lb./sq. in.	195
29.		Final temperature .. ..	° F.	530
30.		Heat per lb. above feed temperature	B.Th.U.	1,178

*E. Flue Gases*

Item 31.	Temperature leaving economiser	..	..	° F.	400
„ 32.	Analysis leaving economiser :—				
	CO <sub>2</sub> .. .. .	..	..	per cent.	10
	O <sub>2</sub> .. .. .	..	..	„ „	9.3
	CO .. .. .	..	..	„ „	Nil

*F. Draught and G. Air*

Items 33 to 38 inclusive	..	..	..	..	Not required for this calculation.
Item 39.	Temperature of air supply to furnace	..	..	° F.	60

## RESULTS

„ 40.	Overall thermal efficiency on gross calorific value of the fuel	..	..	..	per cent.	78.4
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*A. Heat Account Calculated on the Net Calorific Value of the Fuel*

Item 41.	Overall thermal efficiency	..	..	..	per cent.	82.3
„ 42.	Loss due to sensible heat in chimney gases..	..	..	..	„ „	11.9
„ 43.	Loss due to unburnt CO .. .. .	..	..	..	„ „	Nil
„ 44.	Loss due to combustible matter in ashes and clinkers .. .. .	..	..	..	„ „	1.4
„ 45.	Balance :—radiation and other unmeasured losses including grit emission	..	..	..	„ „	4.4
					„ „	100.0

*B. Net Overall Thermal Efficiency*

Not required for this calculation

## CALORIFIC VALUE OF THE COAL

In the B.S. code, the net C.V. of the coal is taken as the basis. The net C.V. may be termed the “available C.V.,” since it ignores the latent heat of the water produced on combustion. Because the net C.V. of the coal is the basis of the method here described, the latent heat of the water in the flue gases is omitted from the calculations.

## LOSSES

These are as shown by :—

(A)	Item 42.	Loss due to sensible heat in chimney gases	..	per cent.	11.9
(B)	„ 43.	Loss due to unburnt CO .. .. .	..	„ „	0
(C)	„ 44.	Loss due to combustible matter in ashes and clinkers .. .. .	..	„ „	1.4
(D)	„ 45.	Balance :—radiation and other unmeasured losses including grit emission	..	„ „	4.4

The sum of these four items, i.e. 17.7 per cent. represents the total losses and therefore  $100 - 17.7 = 82.3$  per cent. represents the overall efficiency of the plant on the net calorific value obtained by the indirect method.

Item 42. If only the theoretical weight of air required for the combustion of the fuel were supplied the chimney losses would be a minimum. This minimum, in the example set out above, would have been 6.4 per cent. ; the actual loss with 10 per cent. CO<sub>2</sub> was 11.9 per cent. Also, if only the theoretical weight of air is supplied the temperature of the gases at the chimney will be reduced. In practice it is impossible to operate with the theoretical weight of air as has been explained in Chapter VI.

The latent heat in the water vapour in the flue gases is ignored, as has just been explained. The sensible heat of the gases depends not only on their temperature but also on their composition, since this affects their specific heat

(Chapter V). As a simplified, though approximate method of calculation, in the B.S. code a method devised by Siegert is used :—

$$\text{Sensible heat in flue gases} = \frac{K (t_2 - t_1)}{\text{per cent. CO}_2 \text{ in flue gas.}}$$

where  $K =$  0.35 for bituminous coal  
0.37 for anthracite  
0.39 for coke.

$t_1$  = temperature of air supply to furnace, °F.

$t_2$  = temperature of gases leaving boiler, economiser or air heater.

It has lately been found that whilst the maximum error in using this simplified formula is  $\pm 1$  per cent. for temperatures of the order of 500° F. commonly met with at economiser outlets, at higher temperatures of the order of 1,000° F. which are by no means unknown, the error rises to  $\pm 3$  per cent. or so. It is therefore recommended wherever possible that the calculation should be made from first principles as set out in Chapter V.

In this example the chimney loss is the largest of the four losses, and this is almost always so in practice. It will be seen that the magnitude of the chimney loss is directly dependent upon the temperature of the leaving flue gases, and on the percentage of CO<sub>2</sub>; a reduction in the gas temperature or an increase in the CO<sub>2</sub> produces a smaller loss.

A low leaving gas temperature can be obtained by :—

- (a) Installing in the first place ample heating surface of boiler and/or economiser, but in determining the size of these sections of the plant consideration has also to be given to the appreciable increase in capital cost that arises when a plant is designed for a very low leaving gas temperature.
- (b) Maintaining the heating surfaces both externally and internally in a clean condition.

With plants that are in good condition and efficiently operated, losses under the headings of Items 43, 44 and 45 of the following approximate order should be expected :—

Item 43.	Loss due to unburnt CO	..	Nil.
	Loss due to combustible matter in ashes and clinkers	.. ..	Not exceeding about 2 per cent. unless the fuel burnt has abnormal characteristics such as high percentage of fines, non-caking qualities, etc., when this loss will be appreciably greater.
45.	Balance—radiation and other unmeasured losses	.. ..	Between 2 and 6 per cent. Where conditions favour grit emission (see Chapter XI) this figure may be very much higher.

## BOILER TEST MEASUREMENTS

In what follows it is assumed that the object of the test is to ascertain what is the best performance that can be expected from a particular boiler plant of small or medium size in use in a factory, or alternatively to ascertain whether any improvement or deterioration has occurred since a previous test.

## DURATION OF TEST

In considering the duration of the test, it is understood that the rate of evaporation is maintained practically constant throughout the period of the test and, once the combustion conditions are set for the kind of fuel being used, it should only be necessary to make minor alterations during the test. The load should, for at least 24 hours before the test commences, be as near as possible to the load carried during the test. Where these requirements cannot be fulfilled, a statement to this effect should be included in the report of the trial and it must be recognised that the recorded efficiency of the plant has been affected accordingly.

The trial should continue until the total fuel consumption per square foot of grate surface is equivalent to not less than 25 lb. for every 1 inch of fire thickness, with a minimum duration of four hours and this period is generally sufficient for the measurement of the other data with the necessary degree of accuracy. Thus, with a 6-inch fire, the trial should continue until  $6 \times 25 = 150$  lb. of fuel have been consumed per square foot of grate surface. At a rate of firing of 30 lb. per square foot, the duration would be at least  $150/30 = 5$  hours.

The plant should be run under test conditions for not less than one hour before and one hour after the trial, during which time sufficient data are to be collected to demonstrate that similar and steady conditions prevailed at the commencement and end of the trial. A four-hour trial plus these two control hours is less liable to error than a six-hour trial without the control periods. The fire should be as far as possible in the same condition (thickness, etc.) at the end of the trial as at the start. With hand fired furnaces, the trial should cover one or more complete cycles from clean fires to clean fires.

With pulverised fuel, direct-fired on the unit system, four hours with two control hours are usually sufficient. Where the pulverised fuel is not fired direct from the pulveriser, either in the unit or central system, a longer period is usually necessary for a correct determination of the fuel consumption. Such duration can only be determined by an examination of the plant and the possible variation of the quantity of fuel contained in the system between the point of weighing the fuel and its delivery to the furnace. The test should be of sufficient length to reduce this variation of the quantity of coal contained in the system to a percentage of the total fuel consumed, consistent with the degree of accuracy aimed at, which should ordinarily be about 1 per cent.

## COAL

The coal used for the test, in so far as circumstances permit, should be the same as that normally used. The amount used should be weighed as indicated in Chapter X.

The coal should be sampled in accordance with B.S. Specification 1017 (see Chapter XXXI) and the calorific value, together with the proximate analysis, determined by a competent chemist. When a staff chemist is not available the services of a firm specialising in this class of work should be sought.

The accuracy of the result of the test is entirely dependent upon the accuracy with which the calorific value of the fuel is determined. This depends upon two factors, (a) proper sampling of the fuel and (b) determination of the calorific value. Any inaccuracy occurring in either of these factors will be directly reflected in the final result. The important part which proper sampling of the fuel plays in the determination of boiler efficiency is not always realised, and it is therefore essential that the instructions for sampling laid down by B.S. 1017 be rigidly followed.

At the same time as the sample for the determination of calorific value is taken, a further sample of at least 20 lb. in weight should be prepared for the purpose of making a sizing analysis; determination of sizing should be undertaken by the chemist. Although details of the coal sizing are not necessary to determine the efficiency at which the plant is operating, they provide information which is essential before any recommendation for improvement in the efficiency can be made. The sizing specification should be drawn up by reference to the B.S.I. standard sieve numbers.

## WATER EVAPORATED

Methods of measuring water and steam by orifice plate, and water by V-notch, are described in Chapter X; other forms of water meter are mentioned in Chapter XXX. All instruments should be calibrated before the test.

If the weight of steam produced is taken as the weight of water fed to the boiler, when measuring this water every care must be taken to see that no water is used for any other purpose, that there is no escape of water from the feed system, no leakage from pump glands or economiser safety valves, and that water is not blown down from the boiler during the test. If the boiler is fitted with a continuous blow-down system, allowance must be made accordingly, or the blow-down must be shut off for the period of the test.

In addition to the methods of measurement given in Chapters X and XXX, water may be weighed directly in tanks placed on weighbridges, this being the best method for small installations; or the water may be measured in calibrated tanks, the volumes at the beginning and ending of the test being corrected for temperature.

The water level in the boiler should be the same at the finish as at the start of the test.

It is frequently necessary to determine the wetness of the steam, especially when there are no superheaters, or when superheaters are installed on only half the boilers.

## TEMPERATURES

The temperatures required for the test are as follows:—

Water entering economiser.

Water entering boiler.

Flue gases leaving the end point of the system (i.e. leaving boiler, economiser or air heater).

Air supply to furnace

Air supply to air heater.

Steam temperature.

The measurement of temperature and the precautions necessary to obtain true readings are discussed in Chapter XXX, and also under "Stratification," Chapter IX. If a superheater is fitted, care must be taken to see that the thermometer records the true temperature of the steam leaving the unit. The thermometer pocket should be placed in a straight length of pipe some 2–4 feet from the superheater stop valve.

When taking the temperature of flue gases, special precautions such as a suction pyrometer need not be used if the temperature is below about 750°–800° F. and if the flue is below ground level or is insulated. Nevertheless, adequate precautions must be taken against the errors mentioned in Chapter XXX if the thermometer can "see" any surfaces markedly cooler than those of the gas. Temperatures should not be taken near the economiser outlet, since they are likely to be vitiated by stratification. If it should be necessary to attempt to take temperatures at that position, special instruments must be



used wherever the thermometer or pyrometer can "see" the economiser tubes (cf. Chapter VIII, "Radiation"). The higher the velocity of the gases at the point of measurement, the more nearly is the measured temperature likely to be accurate.

### FLUE GAS ANALYSIS

It is essential to measure the  $\text{CO}_2$ ,  $\text{O}_2$  and  $\text{CO}$  content of the flue gases. The standard method employs an Orsat apparatus. There are many automatic  $\text{CO}_2$  indicators and recorders, but if they are used it is essential that they be calibrated against an Orsat apparatus immediately prior to the test.

It is advisable to stress again the importance of ensuring that the sample of gas withdrawn into the Orsat is a truly representative average sample. This subject is discussed in Chapters XXX and XXXII.

### ASHES AND RIDDLINGS THROUGH GRATE

The ash should be set aside, allowed to cool and then weighed. A sample should then be taken, the same care being exercised as in taking the sample of fuel. The sample is then sent to the chemist for the determination of the combustible content.

The riddlings should always be re-fired; but if it is not practicable to do so during the test the riddlings should be added to the ash prior to weighing and sampling.

### DRAUGHT GAUGES

Draught gauges are necessary for proper control of operation, but draught readings do not come into efficiency calculations.

### PRESSURE GAUGES

The saturated steam pressure is the pressure of the steam in the boiler drum; superheated steam pressure is measured at the superheater outlet. Bourdon gauges are quite suitable for this measurement, but they should be calibrated against a dead weight pressure tester or against a calibrated master gauge. Any correction necessary should be noted and applied.

### READINGS

It will be found that usually all the readings of the various instruments are reasonably steady when the boiler is tested at a constant rate of evaporation, and therefore readings need only be taken at say 15 or 20 minute intervals. It is convenient to have readings taken by individual observers on foolscap sheets and transferred after each reading to a log sheet on which all readings are entered. This enables a watch to be kept on the effects of any changes and also on unforeseen results. The only exceptions are coal consumption and water evaporated, as it is the total of these for the whole test which matters. It is difficult to segregate coal consumption into accurate hourly periods. At the end of the test, the various readings should be averaged, and corrections applied where necessary.

### POWER FOR AUXILIARIES

Power taken for auxiliaries strictly speaking is a debit against boiler efficiency. It is the general practice not to bring this item into normal efficiency calculations. It is essential to take account of this power when it is desired to arrive at a true value of overall output and efficiency, or when comparing different types of plant. For operating purposes, it will be sufficient to set up a standard

of performance for the auxiliaries and then to make a check test at reasonable intervals to find out whether the performance is deteriorating.

The consumption of steam by boiler auxiliaries should be measured with some accuracy. In particular the high consumption of steam jets is often a serious source of loss, and it is important to discover exactly how much steam is being used for this and other purposes about the boiler plant. The accurate determination of these steam quantities demands measuring accuracy of a fairly high order (see Chapter X). When the consumption of the auxiliaries such as fans, steam jets, and pulverised fuel equipment exceeds 1 per cent. of the heat in the coal, it must be deducted in arriving at the net efficiency.

When electrical power is used the equivalent steam consumption per hour is the input to the motors in kW. multiplied by the steam consumption per kW. hour of the prime mover producing the electricity. This conversion factor will vary over a very wide range according to the size and type of the prime mover, but since, as indicated, the evaluation of the power consumed by auxiliaries is only for comparative purposes, it is convenient to adopt the conversion factor given in the B.S. code, namely, 15,000 B.Th.U. per h.p. or 20,000 B.Th.U. per kW. hour.

For steam-operated auxiliaries the steam consumption can sometimes be measured without undue complications, but frequently this will hardly be practicable, and the maker's figures should then be adopted.

When steam jets are used below the fires either for cooling or for forced draught, the steam consumption can be calculated with reasonable accuracy if the diameter of the throat of the jet, and the steam pressure and temperature in front of the nozzle are known. A suitable formula is given elsewhere (Chapter VI). It may be recalled that the passage of high velocity steam leads to erosion and therefore in practice the steam consumption of nozzles gradually increases as wear proceeds. A watch should be kept on this wear, and as the cost of nozzles is very low, they should be renewed at reasonable intervals to keep down steam consumption.

It is usual to exclude feed pumps from individual boiler auxiliaries for various reasons. If it is considered necessary for any reason to undertake a test of the feed pumps, then this should be carried out by means of a separate efficiency test.

## GENERAL REMARKS

Consideration will reveal that the whole object of boiler testing is to compare the results being obtained with some standard. It is therefore not only desirable to have a standard of performance for the particular plant which can be set by the tests outlined earlier, but also to have detailed particulars of what has been achieved on other plants of a similar type and size.

Comparison of the results of a number of efficiency tests (which must be at the same rating) made from time to time on the same plant may reveal a change in performance. When a reduction in thermal efficiency is suspected or known, it may arise from one or more of a large number of causes. A change in efficiency may be due to a change in the character of the coal used. A loss arising from fouling of the exterior heating surfaces, or due to scaling on the water or steam sides, will be indicated by a rise in the temperature of the flue gases; this rise may be due to fouling over many sections of the boiler, or it may be confined to one section. If, for example, the economiser is at fault, the temperature of the gases entering it will be normal, while the temperature of the gases leaving it will be higher than normal; the temperature of the water leaving the economiser will then be lower than normal. Whether the loss in efficiency is due to internal or external fouling must be ascertained by visual inspection. If a change in efficiency is shown by tests, particular care must be

taken to recheck the instruments used before making any major deductions from the results.

If the temperature of the superheated steam is low this may be brought about by one of three causes :—

- (1) Fouling of the exterior surfaces.

This can be determined by visual inspection.

- (2) Priming.

If water is carried over from the boiler drum into the superheater so that the superheater tubes are used for evaporating as well as superheating, a fall in the steam temperature will inevitably result and the separator in the boiler drum must therefore be examined. If solids are carried over with the steam from the boiler drum into the superheater, scale or sludge will be deposited on the inside of the superheater tubes. If this should occur attention must be given not only to the separator in the drum, but also to the treatment of the feed water (Chapter XV).

- (3) Displacement of baffles.

Visual inspection will indicate whether all baffles are correctly located.

The quality of the combustion as indicated by the gas analysis ( $\text{CO}_2$  and CO content) must be watched carefully as has been indicated in Chapters IV and VI.

If the best results are to be obtained from boiler working it is essential that a number of quantities and qualities should be under constant observation. The necessary observations can be made relatively easily if the boiler is properly equipped with suitable and reliable instruments maintained in good working order. The ideal is so to equip the boiler that it can be operated always under test conditions.

The smallest plant should be provided with some instruments so that the operators are aware of what is occurring. The minimum equipment may be considered to be that which would provide the following information :—

- (1) Measurement of coal burnt over given periods.
- (2) Output of boiler plant in lb. of steam or weight of water evaporated.
- (3) Temperature of steam (if superheaters are fitted).
- (4) Temperature of flue gases.
- (5) Temperature of water entering economiser.
- (6) Temperature of water leaving economiser.
- (7) Draught at base of chimney.
- (8) Draught at economiser inlet.
- (9) Draught in furnace.
- (10)  $\text{CO}_2$  content of flue gases.

With larger plants, equipment on a more generous scale is a wise investment, which if properly used will pay for itself many times over.

Even with small plants, if mechanically fired, the efficiency should average between 70–80 per cent., and should be nearer the higher value than the lower. In large power stations average efficiencies of over 85 per cent. are secured over the year. This is achieved by installing plant which will produce a high efficiency, properly testing it under the best conditions, and then arranging the equipment and operation so that all losses are kept constantly to the minimum value.

Two test results (Tables 70 and 71) are set out in the manner recommended by the British Standard code to serve as a general guide.

TABLE 70. LANCASHIRE BOILER

1. Specified Conditions and Performance						
Item	Evaporation, normal load	..	..	..	..	lb./hr. 9,000
	Pressure of steam	..	..	..	..	lb./sq. in. 230
	Final steam temperature	..	..	..	..	° F. 675
	Initial temperature of feed water	..	..	..	..	° F. 110
	Class of fuel	..	..	..	..	Slack coal.
	Overall thermal efficiency	..	..	..	..	per cent. 76
2. Particulars of Plant						
	Boiler (type) Lancashire 30 feet x 9 feet, heating surface	sq. ft.				1,032
	Superheater (type) downtake, heating surface	..	sq. ft.			282
	Economiser (type) gilled tube, heating surface	..	sq. ft.			1,760
11.	Method of firing	..	..	..	..	Coking stoker.
12.	Grate area	..	..	..	..	sq. ft. 48
13.	Draught plant	..	..	..	..	Motor driven induced draught fan.
3. Data from Observations, Analyses, etc.						
14.	Duration of test	..	..	..	..	hr.
A. Fuel						
15.	Description	..	..	..	..	Slack coal.
	Size: Through 1 in. B.S. Sieve	..	..	..	..	per cent. 100.0
	$\frac{1}{2}$ in.	..	..	..	..	92.4
	$\frac{3}{8}$ in.	..	..	..	..	82.7
	$\frac{1}{4}$ in.	..	..	..	..	66.9
	$\frac{1}{8}$ in.	..	..	..	..	46.7
	$\frac{1}{16}$ in.	..	..	..	..	28.8
16.	Proximate analysis as weighed:—					
	Moisture	..	..	..	..	7.9
	Volatile matter	..	..	..	..	29.8
	Fixed carbon	..	..	..	..	57.0
	Ash	..	..	..	..	5.3
						100.0
17.	Gross calorific value as weighed	..	..	..	B.Th.U./lb.	12,890
18.	Net or lower calorific value as weighed	..	..	..	B.Th.U./lb.	12,370
19.	Fuel fired per hour	..	..	..	lb.	1,359
20.	Fuel fired per hour per square foot of grate area	..	..	..	lb.	28.3
B. Ashes and Clinkers						
21.	Collected per hour	..	..	..	..	lb. 81
22.	Percentage of fuel fired	..	..	..	..	5.9
23.	Percentage of combustible	..	..	..	..	33
C. Water						
24.	Temperature entering economiser	..	..	..	..	° F.
25.	Temperature entering boiler	..	..	..	..	° F. 291
26.	Evaporation per hour	..	..	..	..	lb. 10,327
27.	Evaporation per lb. of fuel as weighed	..	..	..	..	lb. 7.6
D. Steam						
28.	Gauge pressure	..	..	..	..	lb./sq. in. 229
29.	Final temperature	..	..	..	..	° F. 670
30.	Heat per lb. above feed temperature	..	..	..	..	B.Th.U. 1282.5
E. Flue Gases						
31.	Temperature leaving boiler	..	..	..	..	° F. 674
	Temperature leaving economiser	..	..	..	..	° F. 416
32.	Analysis leaving economiser:—					
	CO <sub>2</sub>	..	..	..	..	per cent. 9.1
	CO	..	..	..	..	10.1
F. Draught						
33.	Suction over fire	..	..	..	..	in. w.g. 0.8
35.	Suction leaving economiser	..	..	..	..	in. w.g. 1.9

*G. Air*

Item 39.	Temperature of air supply to furnace .. .. ° F.	82
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## 4. RESULTS

„ 40.	Overall thermal efficiency on gross calorific value of the fuel .. .. per cent.	75.6
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*A. Heat Account Calculated on the Net Calorific Value of the Fuel*

„ 41.	Overall thermal efficiency .. .. per cent.	78.8
„ 42.	Loss due to sensible heat in chimney gases .. .. „ „	12.8
„ 43.	Loss due to unburnt CO .. .. „ „	Nil
„ 44.	Loss due to combustible matter in ashes and clinkers.. „ „	2.3
„ 45.	Balance—radiation, grit emitted, and other unmeasured losses .. .. „ „	6.1
		<hr/> 100.0

*Note.*—The somewhat high loss due to combustible matter in ashes and clinker is attributable to the high percentage of fines in the fuel.

TABLE 71. WATER-TUBE BOILER

## 1. Specified Conditions and Performance

Item 1.	Evaporation, normal load .. .. lb./hr.	25,000
„ 2.	Pressure of steam .. .. lb./sq. in.	160
„ 3.	Final steam temperature .. .. ° F.	550
„ 4.	Initial temperature of feed water .. .. ° F.	150
„ 5.	Class of fuel .. .. Scotch washed nuts.	
„ 6.	Overall thermal efficiency .. .. per cent.	84

## 2. Particulars of Plant

„ 7.	Boiler (type) water-tube, heating surface .. .. sq. ft.	6,182
„ 8.	Superheater (type) integral, heating surface .. .. sq. ft.	1,540
„ 9.	Economiser (type) cast iron vertical tube, heating surface .. .. sq. ft.	2,556
„ 11.	Method of firing .. .. Chain-grate stoker.	
„ 12.	Grate area .. .. sq. ft.	140
„ 13.	Draught plant .. .. Motor driven induced draught fan.	

## 3. Data from Observation, Analyses, etc.

„ 14.	Duration of test .. .. hrs.	6
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*A. Fuel*

„ 15.	Description .. .. Scotch washed D.S. nuts.	
„ 16.	Proximate analysis as weighed :—	
	Moisture .. .. per cent.	10.3
	Volatile matter .. .. „ „	32.5
	Fixed carbon .. .. „ „	48.5
	Ash .. .. „ „	8.7
		<hr/> 100.0
„ 17.	Gross calorific value as weighed .. .. B.Th.U./lb.	11,980
„ 18.	Net or lower calorific value as weighed .. .. B.Th.U./lb.	11,460
„ 19.	Fuel fired per hour .. .. lb.	2,878
„ 20.	Fuel fired per hour per square foot of grate area .. lb.	20.6

*B. Ashes and Clinkers*

„ 21.	Collected per hour .. .. lb.	193
„ 22.	Percentage of fuel fired .. .. „	6.7
„ 23.	Percentage of combustible .. .. „	12.2

*C. Water*

„ 24.	Temperature entering economiser .. .. ° F.	155
„ 25.	Temperature entering boiler .. .. ° F.	222
„ 26.	Evaporation per hour .. .. lb.	24,773
„ 27.	Evaporation per lb. of fuel as weighed .. .. lb.	8.61

*D. Steam*

Item 28.	Gauge pressure .. .. .	lb./sq. in.	166
„ 29.	Final temperature .. .. .	° F.	545
„ 30.	Heat per lb. above feed temperature .. .. .	B.Th.U.	1,176

*E. Flue Gases*

„ 31.	Temperature leaving economiser .. .. .	° F.	284
„ 32.	Analysis leaving economiser:—		
	CO <sub>2</sub> .. .. .	per cent.	10.6
	O <sub>2</sub> .. .. .	„ „	8.6
	CO .. .. .	„ „	Nil

*F. Draught*

„ 33.	Suction over fire .. .. .	in. w.g.	0.3
„ 34.	Suction leaving boiler .. .. .	„ „	0.6
„ 35.	Suction leaving economiser .. .. .	„ „	0.9

*G. Air*

„ 39.	Temperature of air supply to furnace .. .. .	° F.	80
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## 4.

40.	Overall thermal efficiency on gross calorific value of the fuel .. .. .	per cent.	84.5
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*A. Heat Account Calculated on the Net Calorific Value of the Fuel.*

41.	Overall thermal efficiency .. .. .	per cent.	88.4
42.	Loss due to sensible heat in chimney gases .. .. .	„ „	6.7
43.	Loss due to unburnt CO .. .. .	„ „	Nil
44.	Loss due to combustible matter in ashes and clinkers.. .. .	„ „	1.0
45.	Balance—radiation and other unmeasured losses .. .. .	„ „	3.9

100.0

The examples in Tables 70 and 71 are for efficient plants. In Table 72 are other results arranged on different lines, based on the gross C.V. of the coal and comparing the heat distribution of two plants, one of which is efficient and the other not.

TABLE 72

	Efficient plant		Inefficient plant	
	B.Th.U.	Per cent.	B.Th.U.	Per cent.
(1) Heat absorbed by water and steam in boilers, superheaters and economisers ..	9,925	79.40	7,588	60.70
(2) Heat loss due to moisture in coal ..	24	0.19	29	0.23
(3) Heat loss due to water from combustion of hydrogen .. .. .	560	4.48	672	5.38
(4) Heat loss due to moisture in air .. .. .	40	0.32	49	0.39
(5) Heat loss due to dry chimney gas.. ..	1,312	10.50	2,400	19.20
(6) Heat loss due to incomplete combustion of carbon .. .. .	—	—	600	4.80
(7) Heat loss due to unconsumed combustible in ash.. .. .	225	1.80	525	4.20
(8) Heat loss due to unconsumed hydrogen, and hydrocarbons, radiation, and unaccounted for .. .. .	414	3.31	637	5.10
	12,500	100.00	12,500	100.00



- (a) by gases.
- (b) from moisture in coal.
- (c) from moisture in air.
- (d) from hydrogen in coal.
- (e) from unburned gases.
- (f) from steam or water supplied under grates.

Per cent. heat lost in carbon in ashes.

"	"	"	"	in dust.
"	"	"	"	in blow-down.
"	"	"	"	in radiation and convection.
"	"	"	"	unaccounted for.

## EFFICIENCY OF BOILER PLANT

Boiler.

Superheater.

Economiser.

Air heater.

## DRAUGHTS

Induced draught fan.

In smoke chamber (Economic boiler).

In superheater chamber (Economic boiler).

Entering air heater.

Leaving air heater.

Entering economiser.

Leaving economiser.

Side flues (Lancashire boiler).

Over fire.

Forced draught.

Inlet air heater.

Exit air heater.

Under grates.

Secondary air pressure.

## ANALYSIS OF GASES

CO<sub>2</sub> — O<sub>2</sub> — CO:    at boiler exit.  
                               economiser exit.  
                               air heater exit or chimney.

Temperature at combustion chamber.  
                               boiler exit.  
                               economiser exit.  
                               air heater exit or chimney.

## FEED WATER AND AIR DATA

Temperature, feed water economiser inlet.

" " " " " economiser outlet.

Per cent. blow-down.

Temperature, preheated air.

Temperature, air entering air heater.

Temperature, boiler house floor.



**ANALYSIS OF CARBON LOSSES**

Total weight of refuse.  
 Total weight of refuse as per cent. of coal.  
 Per cent. carbon in refuse.  
 Dust ex superheater, economiser, air heaters or grit arresters—lb. per hour.  
 Per cent. carbon in dust.  
 Dust in flue gases—lb. per hour.  
 Per cent. carbon in dust.

**ANALYSIS OF STOKER OPERATION**

Fuel bed thickness.  
 Grate speed.  
 Times fires levelled per hour.  
 Report on clinker formation.  
 Water added to coal—lb. coal.  
 Report on coal segregation.  
 Report on smoke formation.

**POWER REQUIRED BY AUXILIARIES**

Mechanical stoker.  
 Coal elevator.  
 Economiser.  
 Pump.  
 Fans.  
 P.F. mills.

**ANALYSIS OF COAL**

Moisture.  
 Ash.  
 Volatile on dry coal.  
 Volatile—dry ashless coal.  
 Calorific value as received—gross.  
 " " " " —net.  
 Calorific value of dry ashless coal.

**ULTIMATE ANALYSIS OF COAL**

Carbon.  
 Hydrogen.  
 Nitrogen.  
 Sulphur.  
 Oxygen.  
 Caking and swelling properties.  
 Ash melting point (reducing atmosphere).

**SIZE ANALYSIS OF COAL**

0 to  $\frac{1}{8}$  inches.  
 $\frac{1}{8}$  to  $\frac{1}{4}$  "  
 $\frac{1}{4}$  to  $\frac{1}{2}$  "  
 $\frac{1}{2}$  to 1 "  
 1 to  $1\frac{1}{2}$  "  
 $1\frac{1}{2}$  to 2 "

## CHAPTER XVII

### GAS PRODUCERS

Gas producer plant—Nature and Characteristics of fuel—Blast and the admission of steam—Clinker—Channelling—Producer efficiency and heat balances—Operation of producer—Purification of producer gas—Utilisation of producer gas.

**P**RODUCER gas is made by blowing a mixture of air and steam through a bed of hot coal or coke when the combustible in the solid fuel is largely converted into combustible gas, consisting of carbon monoxide and hydrogen accompanied by nitrogen from the air. The resulting gaseous fuel may be used in the crude hot state, when it contains a proportion of tar and dust; or it may be cleaned and distributed in a system of supply mains in a manner similar to town gas. It possesses most of the advantages of gaseous fuel, and in consequence it finds wide application in industrial processes.

This process of complete gasification is carried out usually in vertical cylindrical chambers which may be lined with firebrick or may be formed by an annular jacket containing water from which is raised by heat conduction from the fuel bed the steam required to be added to the blast. The fuel is fed by gravity from a charging hopper on to a hearth or grate, which supports the fuel bed, and provides the tuyeres or points of admission of the blast to the fire. The producer gas is conducted from the producer chamber by an off-take flue to a dust catcher or other cleaning plant (if any is provided) in which the gas may be purified from dust, tar and sulphur prior to distribution.

A clear distinction must be drawn between gas producer plants producing raw hot gas, usually distributed through insulated mains to furnaces and plants in their near vicinity, and plants in which the gas is cooled and freed from deleterious constituents. Cooled and cleaned gas may be produced in suction plants or large pressure units equipped with plant capable of cleaning the gas for distribution in relatively complicated systems of mains and for appreciable distances. The principles of gasification in the producer are the same whatever the conditions of purification.

The principles of the reactions between carbon, air and steam that lead to the formation of producer gas were discussed in Chapters IV and V. From what was there said it will have become evident that producer gas should contain as much combustible gas, CO and  $H_2$ , and as little  $CO_2$  as possible, since  $CO_2$  represents carbon that has already been completely burnt in the producer and is of no further value as a fuel. It is also clearly in the interests of fuel economy to effect the decomposition of the steam to the greatest practicable extent.

These objectives, as will also have been deduced from Chapters IV and V, can be achieved only by

- (1) bringing the air and steam into intimate contact with the carbon and allowing an adequate time of contact; this requires a uniform flow of gas through all parts of the fuel bed,
- (2) by operating with a high temperature, well above  $1,000^\circ C.$ , in the reduction zones.

#### ARRANGEMENT OF PRODUCER PLANT

Fig. 127 is a diagrammatic sketch of a gas producer. A mixture of air and steam is blown in at the base and passes through the air distributor into the ash zone. Since the ash serves further to distribute the blast uniformly it is therefore important that it should be in a good granular condition, free from large masses of clinker.

The air-steam mixture then passes up the fuel bed where the reactions described in Chapter IV occur. The oxidation zone extends to a depth of 4–5 inches above the ash zone. The depths of the other zones are not easily determined and will vary with the depth of the fuel bed, the gasification rate, and the reactivity and size of the fuel. As a rough approximation it may be stated that for a generator with 3 feet 6 inches depth of fuel bed the primary reduction zone and the distillation zone will each be about 1 foot deep, whilst the depth of the secondary reduction zone will be approximately  $1\frac{1}{2}$  feet. The reactions occurring in the several zones are summarised in Figs. 13 and 17.

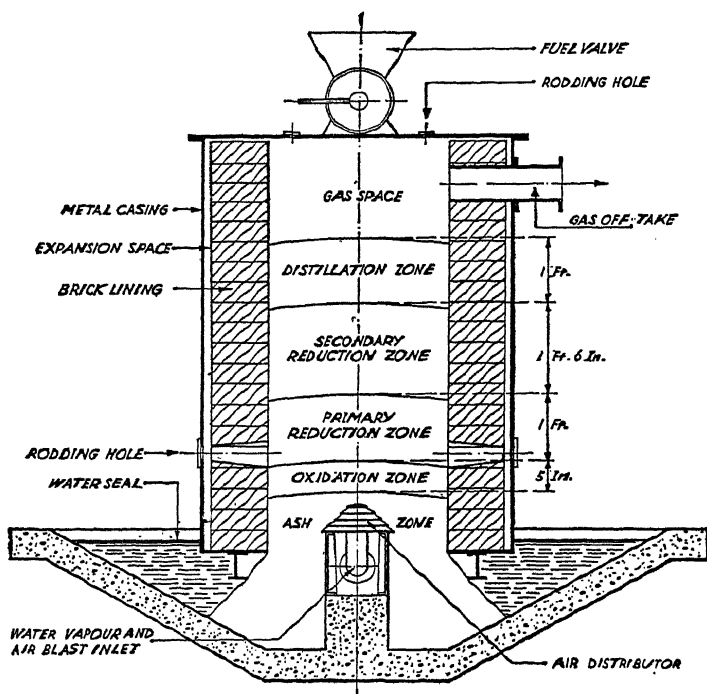


FIG. 127. Diagrammatic sketch of a gas producer.

(Reproduced by permission from "Producer Gas Plant for Industrial Purposes," published by The National Federation of Gas Coke Associations, The British Hard Coke Association, and The South Wales Anthracite and Dry Coal Committee.)

This subdivision into zones is conventional since there is no sharp line of demarcation; the depth of the zones may differ from the figures given.

At the top of the fuel bed is the distillation zone. Here the hot gases come into contact with the freshly charged fuel and the volatile matter is distilled into the gas space above the fuel in the generator. With most anthracites, for example, the volatile matter is converted into permanent gases of which methane is the most valuable on account of its high calorific value. Fuels of high volatile content, including some anthracites, evolve tar vapours in addition to permanent gases. With coke as fuel the distillation zone proper is absent, but some cokes contain "black ends" (i.e. incompletely carbonised portions) which give rise to small quantities of tarry vapours on distillation.

For some purposes the gas can be used in its crude, hot state, containing dust, sulphur compounds and (if made from bituminous, semi-bituminous coal or

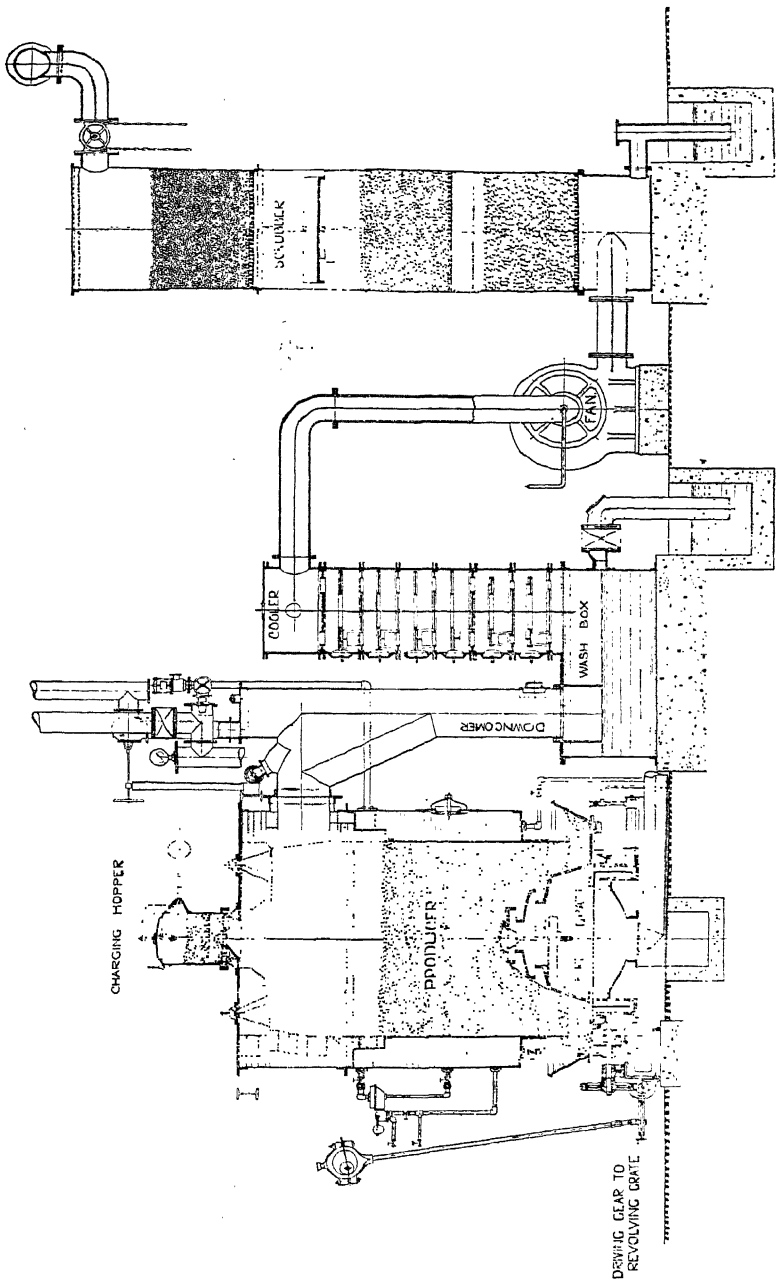


FIG. 128. Diagram of plant for cleaning producer gas.  
(Reproduced by permission from "Producer Gas Plant for Industrial Purposes," published by The National Federation of Gas Coke Associations, The British Hard Coke Association, and the South Wales Anthracite and Dry Coal Committee.)

other high volatile fuel) tar (cf. Chapter II). Use in the crude state conserves the sensible heat of the gas and enables much of its tar to be used. Tar is, however, deposited with dust in the gas mains and these must be cleaned periodically. Except in valves, tar is not usually a serious problem in reasonable flue systems. It does not deposit to a noticeable extent until some considerable distance from the producers. Dust is the main trouble.

For other uses, the gas must be cooled and cleaned from dust and tar. It may also have to be purified from sulphur in gas cleaning plant described later.

Fig. 128 shows a producer with purification plant for cooling the gas and freeing it from tar and dust. Cooling, together with the partial removal of tar and dust, is often effected by direct contact with water.

## FUEL FOR PRODUCERS

The fuels commonly used for gas producers in this country are anthracites, gas cokes and hard cokes, carbonised briquettes, dry steam and non-caking bituminous coals. All these are characterised by the fact that when heated they are non-swelling and non-caking. This ensures that the gases can pass freely between the pieces and come into intimate contact with the fuel surfaces. Further, to avoid packing and consequent uneven distribution of the gases in the generator it is customary to use closely sized fuels. Special producers are designed to operate on smaller-sized fuels, for example, anthracite grains, coke breeze, or on waste products, such as wood chippings, but these plants are generally more expensive both in capital and operating costs, and are only installed when supplies of these fuels are available at attractive prices.

Of the bituminous coals, those which are non-swelling and non-caking (Lessing groups 1 and 2) are the most suitable, but coals of a stronger caking character (groups 3 and 4) can be utilised in suitable mechanical types of plant so that in general a wide variety of coals is used. Suitable coals are to be found in Scotland, Northumberland, Durham, Yorkshire, Notts and Derby, Staffordshire, Warwickshire and South Wales. The best types yield 30–40 per cent. of volatile matter though coals with a lower volatile content are successfully used. Strongly caking coals are the most difficult to gasify, and have to be worked hotter. The output may be considerably reduced by the agglutinating character of the coal unless there is provision of mechanical stirring or levelling arrangements. Coals with an absence of caking power such as the carbonaceous coals of South Wales, which break down in the fuel bed, require, as with strongly caking coals of Durham and other districts, the application of mechanical producers of special type.

## SIZE CHARACTERISTICS

The more uniform in size the fuel the more readily can the producer be operated in a satisfactory manner since any condition which interferes with the passage of the blast through the fuel bed is likely to reduce the quality of the gas and place a limit on the producer capacity. For bituminous coal the most suitable is a closely sized grade, of the order of  $1\frac{1}{4}$ – $\frac{3}{4}$  inch, containing a minimum proportion of fines. Excessive dust not only chokes the fuel bed, but is carried forward and may cause choking of mains and valves. The upper limit of size used in practice is determined by the capacity of the feeding mechanism.

The same principles apply to coke and anthracite. Coke of the size  $1\frac{1}{4}$ – $\frac{3}{8}$  inch has been gasified under extreme conditions at a rate of 60 lb. per square foot per hour. A size as near  $\frac{3}{4}$  inch as practicable is regarded in some large plants as the most desirable. In others 1–2 inches is used. The size of producer fuels is shown in Tables 74 and 75.

TABLE 74. COAL

Rated capacity of producer		Fuel size inches
lb. per hour.	Therms per hour	
40-45	5	$\frac{3}{4}$ -1 $\frac{1}{2}$
170-850	20-100	$\left\{ \begin{array}{l} \frac{1}{2}-\frac{3}{4} \\ \frac{3}{4}-1\frac{1}{2} \\ 1\frac{1}{2}-2 \end{array} \right.$
Over 850	Over 100	$\frac{1}{2}$ -2

TABLE 75. COKE

Types of fuel	Rated capacity		Fuel size inches
	lb. per hour	Therms per hour	
Gas coke ..	45-50	5	$\frac{1}{2}$ -1 $\frac{1}{2}$
	170-850	20-100	1-2
	Over 850	Over 100	1-2
Hard coke ..	45-50	5	$\frac{1}{2}$ -1
	170-850	20-100	$\frac{3}{4}$ -1 $\frac{1}{2}$
	Over 850	Over 100	$\frac{3}{4}$ -1 $\frac{1}{2}$ (as near $\frac{3}{4}$ in. as prac- ticable)

## REFRACTORINESS OF THE ASH

Fuels with ash of a low refractoriness cause clinkering, whereas those producing ash having a high fusion point, of the order of 1,400° C., permit of the use of less steam than when the ash has a low fusion point, a subject which will be discussed later; the producer can be run hotter and a gas of higher calorific value results. Alternatively an ash with a refractoriness below 1,250° C. will require additional steam, and a limit may be put on the gasification rate practicable. The actual proportion of ash is less important if it is non-clinkering, and the producer is not driven at a rate requiring clinker removal. Clinker formation was discussed in detail in Chapter VI, from which it will be seen that ash fusion temperatures as ordinarily determined form only a very general guide.

For anthracite and coke the lowest permissible ash fusion temperature is 1,150° C. determined under reducing conditions.

Too high a content of ash in the fuel is especially undesirable in hand-clinkered producers, and even in mechanical producers may restrict throughput and give rise to operating troubles through uneven gasification.

## SULPHUR

For certain uses the sulphur content of producer fuel must be kept low: below 1 per cent. for acid steel melting, 1.5 per cent. for metal heating, and 2 per cent. for basic steel melting.

## MOISTURE

Where the gas is used in the crude state, excessive moisture even to the extent of 2-3 per cent. above the normal moisture content of the coal is deleterious, and will lower the flame temperature of the gas in the furnace partly by adding to the inerts and partly by heat-absorbing reactions in which the steam participates. This, of course, does not apply when the gas is cooled and cleaned.

Any water added with the fuel will be evaporated in the producer where it will reduce the outlet temperature of the gases; the steam so evaporated must be condensed in the purification system and thus the lower outlet temperature does not mean less work on the cooler.

## THE BLAST

The composition of producer gas depends more upon the proportion of water vapour present in the air blast than upon the type of fuel gasified. This statement applies more particularly to fuels of low volatile matter content, where the volume of gases obtained by distillation is only a small proportion of the total gas made, but even with high-volatile fuels variations of the water vapour-air ratio largely determine the composition and consequently the combustion characteristics of the resulting gas.

The amount of hydrogen present in the gas is the controlling factor and determines the rate at which the gas can be burned without the flame lifting from the burner ports. Where free-flame burners are used it is, therefore, important that the amount of water vapour present in the air blast should be such that the hydrogen content is not less than about 11 per cent.

The blast is formed by blowing a regulated quantity of steam into the air supply. The temperature of the blast is raised to a predetermined figure, at which the air is supposed to be just saturated with water vapour. This is not exactly correct but the blast temperature is a good and simple practical guide. The quantity of steam added to the blast is thus indicated by the blast temperature, which in its turn is controlled by the steam admission valve. Further steam, up to 25 per cent., is derived from the quenching of the clinker in the water seal of the ashpan.

Since considerable importance is attached to the correct proportion of water vapour in the blast, the design of the blast-mixing box, the steam injector or the vaporiser must receive special attention. The essential features are an instrument to measure the temperature of the blast, and ready means of control. Automatic controls which operate the steam valve on the indication of the temperature of the blast are available and should be used whenever practicable. Steam operated turbo-blowers are also used, the exhaust steam being used to saturate the blast.

Table 76, which is an approximate guide and is not to be taken as exact for all steam pressures and degrees of superheat, supplies the data to relate the saturation temperature to the quantity of steam used.

TABLE 76

Blast saturation Temp. ° C. (B.S.T.)	Wt. of steam, lb. per 1,000 cu. ft. of dry air as measured at N.T.P.	Lb. of steam per lb. of dry air	Lb. of steam per lb. coal gasified
50	6.95	0.086	0.26
55	9.28	0.115	0.34
60	12.35	0.153	0.46
65	16.55	0.205	0.61

Studies of the behaviour of steam in gas producers made by W. A. Bone and R. V. Wheeler (*J. Iron and Steel Institute*, 1907, i, p. 126; 1908, ii, p. 206) indicate that up to 55° C. blast saturation temperature the steam is completely decomposed. Their figures are shown in Table 77.

TABLE 77

Steam saturation Temp. of blast ° C.	Lb. of steam in blast per lb. of coal gasified	Per cent. steam decomposed approx.	Lb. of steam decomposed
45	0.20	100	0.20
50	0.21		0.21
55	0.32		0.32
60	0.45	80	0.36
65	0.55		0.44
70	0.80		0.48
75	1.10	50	0.55
80	1.55	40	0.62

Extensive and informative figures have been obtained by Gibson and Gwyther (Table 78) in relation to the gasification of gas coke.

In Bone and Wheeler's trials rather deeper fires were used than in current practice, and the Mond system superheaters were used in circuit. Whilst the actual blast temperature entering the fire was not given it has been stated that in the original Mond system preheats up to 250° C. were obtained. F. Clements's experiments (*J. Iron and Steel Institute*, 1922) showed a distinct relation between the thickness of the firebed and the steam decomposed, and the results differ from those of Bone and Wheeler. It is doubtful if the normal bituminous coal producers decompose steam as efficiently as Bone and Wheeler found. Slow-burning coke producers feeding engines present totally different conditions.

Experimental information from Clements's work may be thus summarised: As the blast saturation for any given fuel bed depth and rate of firing is varied from 45° to 70° C. blast saturation temperature (B.S.T.) the following changes are found to occur in the composition of the gas from bituminous coal.

- (a) The gas composition shows a gradual change between the following ranges, advancing for each constituent from left to right as the blast saturation temperature increases:—

CO <sub>2</sub>	CO	H <sub>2</sub>	CH <sub>4</sub>	N <sub>2</sub>
3-9	30-20	11-20	0.3-3	52-46

- (b) The heating value of the gas drops from 160 to 130 B.Th.U. per cubic foot (measured at 30 inches Hg. 60° F. saturated).
- (c) The volume of gas produced per lb. dry coal increases from 3-10 per cent. according to the type of fuel and the conditions of operation.
- (d) The combined effect of the above two changes is given by a statement of the cold gas efficiency, which is the gross heat of combustion of the gas expressed as a percentage of the initial gross calorific value of the fuel. An optimum value is found when the steam admission to the blast is about 0.3-0.4 lb. of steam per lb. coal. A corresponding blast saturation temperature range is 53°-57° C. Thereafter with increased admission of steam the cold gas efficiency falls appreciably, and at the same time the temperature of the issuing gases becomes reduced.

The optimum blast saturation temperature, which is generally 55°-62° C., will, however, vary with the nature of the fuel. A fusible ash may involve an increased B.S.T. to avoid trouble even at a cost of sacrificing the quality of the gas. Poor quality gas can result from excessive accumulation of ash and clinker just as easily as from too low a fuel bed temperature. Fuels such as anthracite, high-volatile non-caking coals and soft coke can stand a higher B.S.T. (e.g. 60-62° C.) than gas and hard cokes.



TABLE 78. TEST MADE ON 30 H.P. NATIONAL SUCTION GAS PRODUCER  
(Gibson and Gwyther)

Fuel used. Gas coke of analysis:—

C .. .. 81.75 per cent. Ash .. .. 8.3 per cent.  
H<sub>2</sub> .. .. 1.19 " " Calorific value .. .. 12,759 B.Th.U./lb.  
O<sub>2</sub>, N<sub>2</sub> and S .. .. 8.79 " " Size .. ..  $\frac{1}{4}$ – $\frac{1}{2}$  inch.

Number of trial		1	2	3	4	5	6	7
Saturation temperature of air supply, ° F.	..	86.0	106.5	120.0	124.4	132.3	137.6	147.5
Dry fuel burned per hour in lb.	..	28.16	26.00	24.90	26.15	21.70	23.78	29.40
Weight of H <sub>2</sub> O vapour supplied/lb. dry fuel	..	0.1205	0.2170	0.3665	0.3990	0.5225	0.6310	0.850
Water vapour decomposed, per cent.	..	100.0	100.0	100.0	100.0	85.6	72.7	47.0
Composition of gas by volume, per cent.	CO <sub>2</sub>	2.91	3.48	4.73	5.10	5.06	7.64	10.4
	O <sub>2</sub>	0.84	0.74	0.84	0.75	0.57	0.87	0.68
	CO	29.32	27.52	26.10	25.45	25.4	22.8	19.4
	CH <sub>4</sub>	0.20	0.35	0.36	0.28	0.33	0.36	0.41
Calorific value (calc.)* B.Th.U./cu. ft. at 60° F. sat. 30 in. Hg	H <sub>2</sub>	2.61	8.10	11.0	13.10	10.6	11.0	9.64
	N <sub>2</sub>	63.12	59.81	56.97	55.32	58.04	57.32	59.47
Specific gravity (air = 1)*	..	103.7	116.9	121.8	125.7	118.1	111.3	96.6
Volume of air at 32° F. and 29.9 in. Hg per lb. dry fuel, cu. ft.	..	0.95	0.92	0.94	0.87	0.89	0.91	0.94
Thermal efficiency, per cent.	..	56.1	64.0	54.85	53.0	54.9	54.85	54.71
Theoretical flame temperature ° C. (assuming 10 per cent. radiation loss)*	..	61.95	70.25	77.3	78.6	75.0	71.08	58.02
	..	1,635	1,622	1,634	1,643	1,594	1,544	1,429

\* These figures have been added.

It is to be noted from Table 77 that the steam added to the air is not proportional to the B.S.T., but as the B.S.T. rises the steam added becomes proportionately greater.

### PHYSICAL EFFECTS OF STEAM IN PRODUCERS

- (1) Because of its cooling effect steam reduces the temperature at the base of the producer and thus minimises clinker formation. For this reason it has been the practice to increase the steam admission when experiencing difficulty due to clinker formation. This is not necessarily the best practice, because of the effects on the subsequent reactions.
- (2) The passage of undecomposed steam through the producer cools the reaction zones by conveying heat from them to the upper part of the fuel bed. The importance of maintaining a high temperature in the reduction zone has been indicated in Chapter V. Excess steam must be avoided.
- (3) The reactions between steam and carbon absorb heat and the addition of steam in moderate quantities thus serves in two ways to moderate excessive temperatures at the base of the producer which would otherwise cause clinker with even refractory ashes.
- (4) A balance must therefore be struck between the steam admitted (which absorbs heat) and the air admitted (which generates heat).
- (5) Since the reaction between steam and carbon yields essentially a mixture of  $\text{CO} + \text{H}_2$  having a C.V. of 320 B.Th.U. per cubic foot gross, steam admission increases the C.V. of the gas. When the steam admission becomes so great as to cool the reaction zones unduly, both reactions  $\text{CO}_2 + \text{C} = 2\text{CO}$  and  $\text{H}_2\text{O} + \text{C} = \text{CO} + \text{H}_2$  are repressed, with the result that the  $\text{CO}_2$  content of the gas becomes greater and its C.V. falls. The experiments of Gibson and Gwyther just cited illustrate this point.

### CLINKER IN PRODUCERS

Clinker is formed by the fusion and liquefaction of ash in the hot zones of the generator. In the liquid state it is not likely to cause excessive difficulty, for in fact slagging producers have been used, but on reaching the ash zone it is cooled and forms a hard dense material which may be very difficult to break up and remove.

Causes of excessive clinker formation are :—

- (1) Use of high ash content fuels.
- (2) Inefficient or infrequent rodding.
- (3) Use of fuels containing a high percentage of fines.
- (4) An abnormally high rate of gasification.
- (5) The development of hot spots in the fuel bed.
- (6) The use of fuel having too low an ash fusion temperature.

A certain amount of clinker will almost always form on the walls of brick lined producers, due to the straight face of the wall giving an easier passage for the blast than the body of the fire, thus causing a greater tendency to channelling and hot spots.

The firebed must be cleaned at regular intervals, and the clinker must be removed when necessary, but never less frequently than at intervals which allow of an accumulation of more than 10 inches of ash in static producers. The exact period must depend upon the rate of gasification, the character of the ash, and the circumstances which govern uniform removal of ash. The smaller sizes of fuel generally contain higher relative percentages of ash and since they may be gasified more easily, high local temperatures occur, thus causing fusion of the ash. This condition may, to some extent, be mitigated by avoiding the use of "dusty" fuels.

Excessive rates of gasification result in the liberation of increased quantities of ash and the generation of abnormally high temperatures in the oxidation zone. These factors lead to the formation of a greater quantity of clinker.

Hot spots are usually associated with channelling in the fuel bed—notably at the walls—by hanging of the charge. The cores which form in the main combustion zone give rise to channelling. Blast passes through the channels and there results combustion of producer gas in the fuel bed of the generator. The development of a hot spot should therefore be taken to indicate the need for immediate rodding with the object of eliminating channelling.

Difficulties due to clinker may therefore be decreased by ensuring that the producer is not worked above its rated output, by constant attention to the even distribution and correct depth of the fuel bed, by adequate rodding or poking, which ensures the regular removal of any clinker formed before it can build up to excessive proportions, and by attention to the B.S.T.

### CHANNELLING

Channelling through the fuel bed may be the cause of inefficient working since, as explained above, a vertical channel is formed through which a relatively large proportion of the blast will travel. From this arises a local "hot spot," and it leads to clinker formation.

The uprush of gases through a channel gives insufficient time for chemical reactions to be completed and prevents that intimate contact between blast and fuel which is necessary. Consequently channelling causes high  $\text{CO}_2$  content accompanied by a great deal of undecomposed steam, leading to a poor quality of gas.

The remedy for channelling of this type lies in adequate poking.

There is another type of channelling, perhaps better termed irregular blast distribution, due to segregation. If the producer fuel is not closely sized the larger material will tend to roll to the sides of the hopper or producer, and zones (particularly around the edges) will be formed where the fuel bed has a comparatively low resistance. Thus in some experiments made jointly by the British Iron and Steel Federation and B.C.U.R.A., the resistance of a cold producer bed was found to be 4 inches w.g. for a fuel of average size 1.05 inch, and only  $\frac{1}{2}$  inch w.g. for the same flow with a fuel of average size 2.94 inches. Obviously if these two sizes become segregated in a producer, the bulk of the blast will pass up the larger fuel and the effect will be to overload that portion of the producer; poor gas, high  $\text{CO}_2$ , much undecomposed steam and clinker formation will be the result.

One remedy lies in care in handling the fuel and in installing charging arrangements that will obviate segregation. A general remedy is to use fuel of a close size ratio. The size ratio may be thus defined: If the aperture of the smallest test sieve through which 95 per cent. of the coal will pass as the upper limit be  $a$ ; and the aperture of the largest test sieve on which 90 per cent. of the coal will remain as the lower limit be  $b$ , then the size ratio is  $a/b$  and should not be greater than 3 or 4.

### EFFECT OF DEPTH AND CONDITIONS OF FUEL BED

Generally, increasing the depth of the fuel bed results in markedly increasing the amount of steam decomposed. Thick fuel beds are not always so readily controlled as are thinner beds. Irregularities of feeding, and uneven distribution of the blast can produce channels and leakage of blast round the walls of the producer due to the greater resistance of the thick fuel bed. The results are inefficient operation; the percentage of carbon dioxide increases appreciably and the carbon monoxide falls proportionately; and there is a reduction in the hydrogen content of the gas.

## RATE OF GASIFICATION

The rate of gasification is dependent upon the size and type of fuel, the fusibility of the ash, and the degree to which control of the fuel bed is mechanised by the use of mechanical rakes or pokers and grates. In static producers the rates of gasification giving the most consistent operation are of the order of 10–25 lb. per square foot of fuel bed area per hour. Mechanical producers usually operate in the range 30–70 lb. per square foot per hour. Generally at the high rates dust troubles are more frequent, and there is a greater possibility of clinkering troubles.

Essential conditions for the higher rates of gasification for any given design of producer are close grading of the fuel, whether large or small, and suitability of the coal as regards caking character and the refractoriness of the ash.

## PERFORMANCE ON COKE

Coke of the size  $\frac{3}{8}$ – $1\frac{1}{2}$  inches has been gasified satisfactorily at rates of 60 lb. per square foot per hour, though as with coal the lower rates are less conducive to operating troubles, particularly with fuels of low ash refractoriness. Close grading favours the higher rates of gasification. With more than 15 per cent. of fines below  $\frac{1}{8}$  inch screen, troubles are experienced with blow holes, channeling and excessive clinker. The maximum rate of ash removal is about 10 lb. per square foot per hour, though in practice much lower rates are generally operated, e.g.  $2\frac{1}{2}$  lb. per square foot per hour on static producers. The average thickness of the fuel bed is from 3 to 5 feet.

With a coke of suitable size,  $\frac{1}{2}$ –1 inch, excellent gas is made with a blast saturation of 53°–56° C. With larger cokes the blast tends to short circuit through the centre of the fuel bed, causing channels and clinkering, requiring additional steam. The texture of the ash can be controlled by the steam, the following being the usual indications:—

Large excess causes fine mealy ash.

Moderate amount gives pea-sized nodules.

Marked deficiency causes large clinkers.

## MOTIVE POWER FOR BLAST

Where the air and steam supplies are separately controlled, as in producers equipped with fans or other mechanical blowers, it will be obvious that the blast saturation temperature can be maintained at any desired figure but, where steam-jet injectors are used, the adjustment may be rather more difficult. With steam-jet injectors the air-steam ratio depends on the size and shape of the steam discharge orifice, the steam pressure, the design of the blast pipe and the pressure against which the blast is injected.

Variations due to a variable resistance in the fuel bed particularly affect the quantity of steam used when the pressure is supplied by a steam jet. The higher the pressure, the greater is the quantity of steam injected, thus leading to inefficient gasification. Where steam-jet injection is installed, it is particularly important to use fuel of uniform size which remains uniform over successive deliveries. In a great many plants, particularly in bituminous producers, the ash bed grading is more important than the fuel grading. This can be controlled by the blast temperature and the technique of cleaning. A further method of supplying the blast is the application of the turbo blower, that is to say, a steam turbine drive operating a fan. The exhaust steam from the turbine is used to saturate the blast. Considerable improvement in blast conditions, and economy of steam has been claimed for this method of operation. ♣

## EFFICIENCY OF A PRODUCER

According to the British Standard method of testing gas producers (B.S. 995), the efficiency of a producer plant can be expressed in one of three ways :—

(i) Hot gas efficiency

$$= \frac{\text{Potential + sensible heat of gas per ton of dry fuel}}{\text{Total heat per ton of dry fuel}} \times 100$$

(ii) Cold gas efficiency

$$= \frac{\text{Potential heat of gas per ton of dry fuel}}{\text{Total heat per ton of dry fuel}} \times 100$$

(iii) Comprehensive efficiency

$$= \frac{\text{Potential + sensible heat of gas per ton of dry fuel + thermal equivalent of surplus steam per ton of dry fuel}}{\text{Total heat per ton of dry fuel + thermal equivalent of make-up steam per ton of dry fuel + thermal equivalent of energy consumed per ton of dry fuel}} \times 100$$

Each efficiency can be expressed as desired,

- (a) Taking gross calorific values for potential heat of fuel and gas.
- (b) Taking net calorific values for potential heat of fuel and gas.

## HEAT BALANCE

The cold gas efficiency in the generation of producer gas usually ranges between 63–80 per cent. Radiation and cinder loss is not over 9 per cent. Consequently the sensible heat in the gas is from 11–28 per cent. of the total heat input. About 97–98 per cent. of the total heat input usually comes from the fuel itself, the balance being the heat in the steam contained in the blast.

## CONDITIONS FOR EFFICIENT OPERATION OF PRODUCERS

The following summary of conditions favourable for efficient gasification may help to crystallise in a practical way much of the foregoing information.

- (a) The most important individual factor in efficient gasification in blast producers operating on bituminous fuel is the selection of a suitable coal. The relative properties of coal vary very considerably, not only as between one coalfield and another, but also from seam to seam in a given district, and sometimes even in a particular colliery. Broadly, the properties which are desirable are reasonable uniformity of size, low or moderate caking power and a relatively refractory or infusible ash having a minimum tendency to fuse into clinker or on to the brick-work lining of the producer. Some operators prefer that the coal used shall have a moderate caking power in order to avoid disintegration of the fuel in the producer. A useful survey of the suitability of British fuels for this purpose is given in the *Fuel Economy Review* for 1942, page 4.
- (b) A correct depth of fuel bed requires that the grate or tuyeres should be covered with a sufficient but not excessive depth of well-burnt ashes, this depth in static producers being probably not less than 6 inches or more than 12 inches above the highest point at which the blast enters the producer, and that this ash bed should support an adequate depth (usually falling within the limits 2 feet 6 inches to 3 feet 6 inches) of fuel in progressive stages of carbonisation and combustion; both the ash layer and the fuel bed should be of reasonably uniform texture and depth. Wide variations from the above conditions occur in gas machines, depending on the design and operation of the machine.

- (c) Efficient distribution of the blast requires that there should be an equable flow throughout the entire cross-sectional area of the fuel bed, resulting in a uniform rate of gasification. Difficulties in this respect usually arise either from the formation of clinker fused on to the producer lining and thereby restricting the effective area of gasification or from the formation of a hard and relatively impenetrable central core of coke resulting from the carbonising of fines in the fuel and undue concentrations of fine ash locally in the ash bed.
- (d) The blast saturation temperature should be maintained constantly at the figure best suited to the requirements of the fuel and plant and should rarely exceed  $60^{\circ}\text{C}$ . and never  $62^{\circ}\text{C}$ . unless there are exceptional reasons to the contrary.
- (e) The gas pressure in the producer and mains should be maintained at a constant and moderately low figure, which, usually, should not exceed about 1 inch water gauge. An excessive gas pressure causes a material increase in the gas leakage from the poker holes and fuel charging mechanism and, in producers operated by a steam-jet air injector, also tends to increase the difficulty of maintaining a low blast saturation temperature.
- (f) The maintenance of a reasonably low gas exit temperature is desirable as indicating an equable rate of gasification over the entire fuel bed area and a correct depth of fuel; whilst, on the other hand, an abnormally high exit temperature always suggests an impoverishment in the quality of the gas, and sometimes combustion of gas above the fuel bed. A low gas temperature occurs with excessively high blast saturation temperature, and low rates of gasification.

### PRACTICAL OPERATION OF A PRODUCER

#### FUEL CHARGING

Hopper charging with manual operation is the usual practice; mechanical feeding is a general feature of mechanised units. The hopper has a gas-tight lid and is provided with a bottom bell or valve capable of being raised or lowered from the outside to admit the charge of fuel into the hopper and thence to the generator chamber. Some types of mechanical feed are provided with a counter mechanism to indicate the rate of feed. The uniform distribution of the fuel may be assisted by the use of some form of fuel-distributing device which must be adequately protected against damage by heat from the hot gas.

#### LEVELLING OR POKING MECHANISMS

In rotary producers, in which the body of the machine revolves, the levelling of the fuel bed may be effected by a leveller or rake. In both mechanical and static producers oscillating pokers are also used to ensure even distribution of the fuel. In all these types water cooling is used and accordingly detection of water leakage is important. Further it may be necessary to overcome erosion of the wearing parts by the use of special materials.

#### THE GRATE OR BLAST DISTRIBUTOR AND ASHING PLOUGHS

Grates vary from a simple construction of straight grate bars to complex mechanical contrivances. Where the grate or "mushroom" is made to rotate, a helical contour is given to the tuyere rings to cause the ash to be removed radially from the ash bed. In the simpler stationary producers the ash is removed from a water-sealed trough by long-handled shovels or from the ashing doors in dry-bottom producers. Mechanical grates may be provided with ashing

ploughs the use of which at stated intervals determined from the indications of the ash-level test, described later, ensures the correct thickness of ash bed above the grate.

Observation of the condition of the clinker removed will ensure the correct use of steam in the blast. An ultimate feature of all gas producer operation is that the operator becomes concerned mainly with clinker trouble. Continuous mechanical extraction of ash gives a gas of more constant composition, and is stated to increase the overall efficiency by about 5 per cent.

## LIGHTING UP AND CLOSING DOWN PLANT

For lighting up it is usual to build a fire of kindling material such as shavings and timber, on a bed of suitable ash. The kindling may then be set alight, or alternatively, a layer of 1 foot of fuel may be placed on the top first, and the shavings then lighted. When the bed is alight, the fuel is fed gradually, while the lean gases escape at the blow-off pipe. The newly lighted fire is blown by a gentle blast.

In large coke-fired producers, the kindling materials are built into robust beds in which are contained the full burden of coke. The lighting is then effected from tinder boxes.

Explosions both in producers and furnaces during starting-up can be avoided if it is recognised that an explosion can only occur when a sufficiently rich air-gas mixture has accumulated. The ignition temperature of producer gas (Chapter V) is about 600° C. A furnace can be more safely "gassed" with lean gas.

When it is necessary, as in shutting down, to purge the producer and furnaces, purging should be continued until the air has been dislodged from pockets. With clean gas a satisfactory and simple test is to observe the condition of the flame at a test burner. This method is quite safe provided a proper test burner is used. The flame in the burner shows a double cone, similar to that of a lightly aerated bunsen burner, while there is air in the gas. As soon as the inner cone disappears, the gas may be safely used. With dirty gas a cautious trial at a poke hole will show clearly enough what is the composition of the gases. In burning out or shutting down, the main safeguard is to avoid the possibility of the formation of explosive mixtures.

## RUNNING AND CONTROL

In order to put the principles which have been discussed to practical effect, care in operation should be devoted to three essentials. These are :—

- (1) Control of the steam supply.
- (2) Maintenance of the correct depth of fuel bed by attention to the feed. This is usually indicated by maintaining a constant gas exit temperature. The attention to the fuel bed also involves cleaning off the walls any clinker that may form there.
- (3) Efficient removal of the ash, including judicious movement or poking of the fuel bed to prevent the formation of channels and to break up any masses of partly carbonised fuel that may form.

### (1) CONTROL OF STEAM SUPPLY

The blast saturation temperature should be maintained constantly at the figure best suited to the requirements of the fuel, the plant and the conditions of gasification. It should generally be kept below 60° C. Higher rates of gasification may require a somewhat higher temperature to be carried in the blast.

## (2) CORRECT DEPTH OF FUEL BED

The grate or tuyere should be covered with sufficient but not excessive depth of ashes, preferably between 6 and 12 inches above the highest point at which the blast enters the fuel bed. The hot zone and the distillation zone should also be kept within regular limits by means of tests with the test rods. These should be  $\frac{3}{4}$  inch diameter and long enough to reach to the top of the blast tuyere. They are inserted vertically through the poking holes and left in the fuel bed for a fixed interval of time, usually one to two minutes, according to the class of fuel and conditions, and then withdrawn. A red-hot length on the rod will indicate the position of the oxidising zone. This should be sharply defined and only a few inches long. The presence of two or more such zones is an immediate indication of poor fuel bed conditions.

A second rod with a small L-bend at the end may be inserted afterwards to locate the position of the surface of the fuel bed in producers not filled up to the top of the generator.

A graduated rod painted to mark the correct limits should be used to determine the exact measure of the various levels. These should be recorded in the log book or marked in chalk on a board set up on the control stage, together with the times and degree of ash removal, e.g. with mechanical grates the number of revolutions during which the plough has been operating.

The fuel must be fed at a regular rate, otherwise in bituminous coal producers the gas will be of irregular quality. The spread of the coal must be watched, and if there are no mechanical aids to spreading such as deflection plates, the operator may control the spread of the fuel by varying the load of each charge.

One of the hardest and consequently most shirked tasks in producer operation is the poking of the fuel. A light rod may be used for the purpose of levelling.

## (3) CLINKERING AND REMOVING ASHES

The avoidance of excessive clinkering is mainly effected by steam control, but if clinker has once begun to form, it may have to be broken up by the use of the heavy poker, particularly if it is building up on the wall of the producer. Clinkering may arise from the formation of channels, and these may have to be filled by poking, but in carrying out this necessary action care should be taken to avoid too much disturbance of the fuel bed.

## OBSERVATIONS OF THE GAS TEMPERATURE

The temperature of the gas should be measured in the main and the off-take of the producer. The most reliable means is the use of a base metal thermocouple. The use of a recorder for the purpose is one of the most effective means of control, since for a given fuel and fixed rate of gasification and blast saturation temperature the temperature of the issuing gas should remain constant. By considering the temperature of the gas in relation to that of the blast and the fuel bed levels as shown by the rod test, a reliable judgment of the condition of the fuel bed can be formed.

The temperature of the hot gas will generally be for bituminous coal between 500°–750° C. and for coke and anthracite, 300°–600° C. Gas temperature variations may result from channelling in the fuel bed or leakage of air and burning of the producer gas; an unusually low gas temperature may be due amongst other causes to a high moisture content in the incoming fuel.

## GAS QUALITY

The best method of checking gas quality is to make an analysis of the gas. The interpretation of the analysis will be apparent from the earlier discussion



(cf. also Chapter II). Judging the quality of the gas by observation of its character as it leaves the poking holes requires experience tempered with knowledge derived from gas analyses. If, however, the gas from bituminous coal ignites at the poking hole, or appears to be smoky, either the fire is too shallow or channelling may be occurring.

Too high a blast saturation temperature is apparent when the gas appears to be dull and cold, and moisture is deposited if it is brought into contact with the hand. The amount and kind of volatile matter which is yielded from a particular fuel will influence the colour of the gas and therefore the conclusions derivable.

## CONTROL

Among essential instruments are those for measuring the temperature of the blast and the gas at the off-take. The maintenance of the pressure of the gas at a satisfactory and constant value requires either the use of automatic control or a readily visible pressure gauge, placed in a prominent and convenient position. The installation of modern instruments is a powerful means of attaining efficiency of operation in all classes of plant, and should be looked upon by the engineer as a profitable form of scientific control. The metering of coal, air, gas and steam can be readily and efficiently applied.

The most desirable instruments are :—

- (a) Exit gas temperature recorder.
- (b) Blast saturation temperature recorder.
- (c) Blast pressure indicator.
- (d) Gas pressure indicator and controller.
- (e) On plants blown by live steam, a steam pressure gauge.
- (f) On plants blown by electric fans, an air flow indicator.

This equipment is sufficient to disclose all that need be known for the day-to-day operation of most plants. On large batteries of producers more elaborate equipment may be installed in control panels, and the refinement of automatic control used.

The blast saturation temperature should be measured by a thermometer placed immediately before any preheat is given to the air-steam blast and with the bulb immersed to a depth at least one-third of the diameter of the main.

The pressure conditions on the plant are of great importance as fluctuations in these readings give indications of faults which would otherwise be unsuspected.

The actual pressure readings will depend upon the size and the design of plant and the characteristics of the fan employed, but a gradual building up of pressures (in clean gas units) may in general be taken to indicate the need for inspection of either the washer-cooler or dry scrubber, and gives an indication that a fresh charge of the scrubbing medium is required.

On producers equipped with gas-cleaning provision, four gauges of the ordinary water U-tube type are suggested and these should be placed in the following positions :—

- (a) Washer-cooler inlet.
- (b) Washer-cooler outlet.
- (c) Dry scrubber outlet.
- (d) Fan outlet.

## GAS ANALYSIS

It is recommended that analysis of the gas should be made from time to time, preferably at least once a day. A suitable gas analysis apparatus for works purposes is the Orsat, though this is primarily of service for rough works

control. Chapters XXX and XXXII should be consulted for further information.

## THE PURIFICATION OF PRODUCER GAS

The purification of bituminous gas in all but the smallest quantities presents a more difficult problem than that of coke and anthracite gas and is more rarely undertaken. For coke and anthracite gas the usual equipment consists of a washer-cooler in which the gas is cooled and freed from the greater part of its dust content and almost all of any traces of tar by passing upward through a tower charged with coke and sprayed with water. This may be followed by a filter box packed with wood chippings, wood wool, sisal fibre or some other filtering medium, to remove the last remaining dust. This box may also be charged with iron oxide to remove hydrogen sulphide. A gas-washing fan or other mechanical washer may be substituted for the filter box.

In large plants operating on coke the cooling tower is followed by one or two centrifugal disintegrators, containing rotating blades, which are supplied with a stream of water fed to the machine. Violent turbulence between gas and water is produced, and the latter is separated from the gas by centrifugal action. The "disintegrator," intended mainly to remove dust, is followed by a "spray separator."

The washer-cooler is generally packed with graded coke, 1-2 inches in size, over which water is sprayed amounting to some 25 gallons per 1,000 cubic feet of gas made, or some 4,000 gallons per ton of fuel gasified. This water may be run to waste if water is plentiful, or it may be circulated through a cooling pond. Where the water is re-circulated it must be remembered that it will become acid through absorption of oxides of sulphur from the gas and it must be neutralised with lime to avoid corrosion. The presence of sulphur dioxide in coke producer gas has been established, and evidence of serious corrosion adduced where the alkalinity of the wash water has not been watched.

The loss of pressure across the washer-cooler is kept as low as possible by using a suitable packing. If it becomes high the inference is that a blockage is occurring and the scrubber must be repacked with fresh coke of the right size.

The quantity of water used can be best regulated by placing a thermometer in the outlet gas main and adjusting the water consumption to give the required temperature. To avoid difficulties in the cleaning plant the following precautions are suggested :—

- (1) Examine the water sprays regularly.
- (2) (a) Inspect the water overflow at the seal pots and make sure there is no blockage caused by accumulation of sludge.  
(b) In frosty weather take steps to prevent freezing.
- (3) Prevent wastage by using just sufficient water to cool and clean the gas effectively.
- (4) Take records of the pressures at the inlet and outlet connections of both the washer-cooler and the filter.

Difficulties sometimes occur with gumming or sticking of valves and suction fans on producer plants. This is frequently due to traces of tar escaping from the tar scrubber, the presence of which may be shown by allowing the gas to play on a wetted filter paper. If tar is absent, the cause may be the formation of gum by the action of small traces of nitric oxide formed in the producer on certain unsaturated hydrocarbons (e.g. butadiene and cyclo-pentadiene) formed during the distillation of the coal. This type of deposit would not occur with coke. Some cokes, as has been mentioned before, will contain traces of tarry matter, and it is not good practice to do without the tar removal apparatus when using coke as a producer fuel.

Another cause may be deposits due to salt. The salts are derived from the coal, or they may be equally derived from coke which has been quenched with water containing salts. The use of brackish water in the seals of the producers or in the scrubbers, etc., of the purification plant may also give rise to salt deposits.

In most installations it is not considered necessary to remove sulphur, and unpurified gas containing up to about 70 grains of sulphur per 100 cubic feet is fed direct to the appliances. This practice is justified providing :—

- (a) The sulphur compounds in the gas do not adversely affect the material being processed.
- (b) All appliances using large volumes of gas are fitted with flues which carry the products of combustion outside the building.
- (c) Adequate ventilation is provided where flueless apparatus is in use.

Most fuels recommended for use in gas producers contain not more than 2 per cent. of sulphur and on gasification some 75 per cent. of this appears in the gas as a mixture of hydrogen sulphide and organic sulphur compounds. The hydrogen sulphide content of producer gas is generally between 35 and 60 grains per 100 cubic feet according to the sulphur content of the fuel. With high sulphur fuels sulphur contents as high as 140–150 grains may quite well be encountered.

Where it is found necessary to remove hydrogen sulphide the oxide purification system is used which is already well-established in gas works. The producer gas is passed through layers of solid iron oxide specially prepared for the purpose which is thereby converted into iron sulphide. The purification material is removed from the boxes from time to time, when the air serves to reconvert the sulphide into oxide with liberation of free sulphur. The oxide can then be used again until it contains over 50 per cent. of sulphur when it can be sold for the manufacture of sulphuric acid.

Where producer gas containing tar has to be cleaned it is of fundamental importance that the greater part of the dust is removed before any of the tar vapour is allowed to condense. This is accomplished by providing a large and well insulated dust catcher before the cooler. The most satisfactory design of dust catcher is that involving the principle of the cyclone separator. That is to say, the gas enters the dust catcher towards the top at its widest part. It moves horizontally and tangentially to the inner surface of the dust catcher and leaves through a central off-take passing down the axis of the chamber. The use of baffles has been shown to be of little service, whilst unduly increasing the resistance to flow of the gas. From the dust catcher onward the plant does not differ from those for coke and anthracite gas except that the washer-cooler tower must not be packed with coke. In some plants electrostatic precipitators are substituted for the filter boxes for removal of fine dust and tar fog with considerable success. When cleaning bituminous fuel gas, arrangements must be made to separate the cooling water from liquid tar and to dispose of the tar.

## UTILISATION OF PRODUCER GAS

Producer gas may be utilised in two ways. It may be decided to operate furnaces or gas engines on producer gas as a regular practice and in this event the whole installation would be designed to suit the producer gas. On the other hand, producer gas may be used only in an emergency ; producers are installed at many works to-day where town gas is used in order to provide a standby for emergency purposes if war damage should affect the supply of town gas.

Where producer gas is to be used for emergency purposes care should be taken in advance to ensure that the conditions under which it is to be used are known and can be fulfilled. The essential conditions are :—

- (1) That the gas mains in the works under the conditions of pressure to be applied are large enough to convey the necessary quantity of producer gas to the furnace.
- (2) That the arrangements at the furnace are suitable for burning producer gas.

For all practical purposes, and at low pressures, the pressure required to convey gas through a main is given by Pole's formula (Chapter IX).

where  $Q$  = gas discharged through the pipe in cubic feet per hour.

$d$  = diameter of pipe or main in inches.

$p$  = loss of pressure in pipe between inlet and outlet, in inches w.g.

$s$  = specific gravity of gas (air = 1).

$L$  = length of pipe (yards) 5 feet being added to the measured length for each sharp elbow or tee, and 2 feet for each 90° streamlined bend.

In making the calculation it is to be remembered that the quantity of producer gas to be used will be slightly greater (10 per cent. more would be allowed) than would be given by the expression :

$$\frac{\text{Maximum quantity of town gas (cubic feet per hour)} \times (\text{C.V. of town gas})}{\text{C.V. of producer gas}}$$

Thus if a works using 5,000 cubic feet per hour of town gas of 500 B.Th.U. per cubic foot has to change over to producer gas of 125 B.Th.U., the quantity of gas now required is  $5,000 \times 500/125 = 20,000$  cubic feet per hour. To this should be added 10 per cent., making 22,000 cubic feet per hour as a margin for the slightly greater amount of producer gas that may be necessary.

To carry the example further, if the existing gas pipes were 5 inches internal diameter and 100 yards effective length taking 5,000 cubic feet per hour of gas of specific gravity 0.4, the fall in pressure would be practically 2/10 inches (0.2 inch) w.g. To carry the equivalent quantity of producer gas (specific gravity 0.9) through the same mains, would require a pressure drop of 7.6 inches w.g.—an increase of 38 times.

If this does not provide sufficient pressure at the burners, a booster, which can be by-passed when on town gas, must be installed in the system, preferably near the gas plant.

In practice, just as the delivery pressure of town gas is frequently of the order of 2-3 inches w.g., so with producer gas plants, it is the practice to ensure a minimum delivery pressure much higher than the figure calculated above, since there are other resistances than those shown in the simple example cited.

The size of mains thus calculated will be reasonably correct for purified gas, though generally a small allowance should be made for unexpected resistances by making the mains a little larger than indicated by this calculation. For unpurified gas a liberal allowance should be made. The attack by unpurified gas on iron services and copper alloy fittings, particularly at the elevated temperatures frequently to be found around furnaces may be serious.

Whether the pressure obtained at the burners will be enough can be calculated from the flow of gas through burner orifices. This is given by the formula :—

$$Q = 1,658.5 a k \sqrt{p/s}$$

where  $Q$  = gas discharge, cubic feet per hour.

$a$  = area of orifice in square inches.

$k$  = coefficient of discharge, normally 0.75 for gas burners.

$p$  = initial gas pressure at orifice, inches w.g.

$s$  = specific gravity of gas (air = 1).

The pressure at the outlet of the producer gas cleaning plant is known and is generally between 12 inches and 20 inches w.g. Let it be supposed to be 15 inches w.g. The pressure loss in the mains is 7.6, making  $(15 - 7.6) = 7.4$  inches effective pressure available at the burners.

If the coal gas were fed to the furnaces through 10 similar burners of  $\frac{3}{16}$ -inch bore, it could be replaced by 2,200 cubic feet per hour of producer gas through each burner, and the pressure then required at the burners would be given by

$$2,200 = 1,658.5 \times 0.249 \times 0.75 \sqrt{p/0.9}$$

whence  $p = 45$  inches w.g.

It would clearly be necessary to install a booster to overcome part of the pressure drop in the mains, and to enlarge the burner orifices so that the required amount of gas can be passed by some 10–15 inches w.g. pressure. In the foregoing example the booster should be capable of increasing the effective pressure at the burners to, say, 12 inches w.g., and the burners should be enlarged to  $13/16$  inches internal diameter; under these conditions each burner will pass 2,350 cubic feet per hour.

Given a pressure loss of 7.6 inches w.g. and a gas pressure at the producer outlet of 25 inches w.g., there would be left a pressure of the order of 17 inches w.g. on the burners, so that the mere drilling out of the furnace jets would be quite sufficient to ensure satisfactory operation on producer gas without installing a booster, and it is in fact for this very reason that the large majority of makers of these producers have adopted the higher gas pressure referred to above. Lower outlet pressures are only adopted in small units which can be mounted near the point of service.

## FURNACE HEATING BY PRODUCER GAS

In most furnaces combustion takes place in refractory burner throats. Because of this restriction of the gases and the close proximity of hot brickwork to the burner no difficulties are experienced with flame stability, and the relative proportions of hydrogen and carbon monoxide present in the gas are immaterial. The composition of the gas, however, determines the setting of the primary and secondary aeration controls and it is therefore important that the composition should be kept as constant as possible. Much of the efficiency of utilisation depends upon the correct setting of the primary air controls and the secondary air dampers. On most furnaces operating on low pressure producer gas at temperatures up to about  $1,000^{\circ}\text{C}$ ., it will generally be found advisable to close the air slide at the injector and thus use no primary air. The secondary air dampers should then be adjusted so that the  $\text{CO}_2$  content of the flue gases is between 17 and 19 per cent., at which value the atmosphere in the furnace will be oxidising.

Furnaces designed for town gas and working at temperatures up to  $1,000^{\circ}\text{C}$ . will give an equal performance on producer gas provided the heat input is the same, but the heating-up time may be 5 per cent. longer with producer gas.

The maximum temperature which can be attained by any furnace depends upon the flame temperature of the gas used, and the intensity of mixing of gas and air. For producer gas the theoretical flame temperature is approximately  $1,700^{\circ}\text{C}$ . (Chapter V), whereas for town gas the value is some  $400^{\circ}\text{C}$ . higher. It has been found that on a small high-speed furnace without preheating the gas and air a temperature of  $1,325^{\circ}\text{C}$ . can be obtained with producer gas. Above this temperature preheating of air and gas is essential.

The efficiency of some high temperature furnaces heated with producer gas is considerably lower than that of similar furnaces heated with town gas. Comparative tests made on a town gas high-speed steel furnace converted to use producer gas show that the heat input with producer gas is about 25 per cent.

greater than that required with town gas. The heating-up rate is also slower when using producer gas.

Whilst the example of the small high-speed furnace is an experimental fact it is not the practice to operate such furnaces without some degree of preheat, for the reason that in order to avoid an oxidising flame an excess of gas is necessary. A simple form of preheater applied to tool hardening furnaces is effective in giving the required flame intensity. In general furnace practice a much lower minimum limit of temperature of operation on cold gas and air is used. A temperature of  $1,050^{\circ}\text{C.}$  can readily be obtained, but higher temperatures must depend entirely on conditions and particularly on the intensity of mixing and conditions in the combustion chamber. When considering the application of preheat to producer gas it must be realised that preheating of both gas and air is readily practicable, and is facilitated by the fact that the relative volumes of gas and air are very nearly the same. Their densities are also not greatly different so that mixing conditions in the burner under preheat are not likely to interfere seriously with the stability of the flame. The loss of sensible heat in the waste products of combustion for similar furnace temperatures is higher with producer gas than with richer gases, and this conduces to the efficiency of the preheater.

## EFFECT OF HYDROGEN

The linear velocity of the ignition wave of hydrogen-air mixtures is immensely greater than for other gases (cf. Chapter V).

Many industrial and domestic gas appliances are heated by burners where the gas flames burn freely in air and do not impinge on any solid object. These are known as "free flame" burners and the stability is particularly susceptible to changes in the hydrogen content of the gas.

Experiments have been shown that with these burners gases with a low hydrogen content tend to lift off the burner head unless fed with a low linear velocity. What this limiting velocity in the burner head should be can be deduced from the following experiments (Table 79) carried out by the Gas Light and Coke Co., London, which were conducted on a range of orifices between  $\frac{1}{8}$  and  $\frac{3}{4}$  inch diameter.

TABLE 79

	Producer gas				Coal gas
	2	7	12	15	50
Percentage hydrogen in gas .. ..	2	7	12	15	50
Mean velocity of gas in tube in ft./sec. at which the flame lifted from the burner ..	8	17	48	78	134

## LITERATURE

Among recent books on the subject of gas producers particular reference is here made to two small practical works, to which the authors of this chapter have been indebted for some of the material here published :—

"The Efficient Working of Gas Producer Plant," being Fuel Economy Monograph, No. 3, issued by the Technical Department of the Federation of British Industries, 1941. (Price 2s. 8d.)

"Producer Gas Plant for Industrial Purposes," issued jointly, in 1942, by the National Federation of Gas Coke Associations, the British Hard Coke Association, and the South Wales Anthracite and Dry Coal Committee. (Price 5s.)

## CHAPTER XVIII

## FURNACES

The selection of furnaces—Furnace atmospheres—Combustion and heat transmission—Sources of heat loss—Thermal efficiency—Recovery of waste heat—Insulation—The heat losses associated with continuous and intermittent operation—Radiation losses through openings—Other losses—Scientific control—Instruments—Records—Practical hints to operators of furnaces fired by coal, coke and gas, pulverised coal and oil.

**T**HE first function of an industrial furnace is to produce a satisfactory product and other considerations must be subordinated to this. In the past this principle has so prevailed that the study of the fuel efficiency of many furnaces has been neglected. When the cost of fuel is only a small proportion of the total manufacturing costs there is a disposition to regard fuel conservation as unimportant; only in those operations where the cost of fuel is important in relation to the total cost is there the incentive to watch fuel economy closely.

The efficient use of industrial furnaces, therefore, requires the consideration of many factors quite distinct from economy in fuel. Over the many uses to which furnaces are put conditions may be expected to vary widely. Certain principles are, however, common to all furnaces and it is to these principles that attention is primarily directed in this chapter.

The fundamental principles of combustion and heat transmission which have been discussed in earlier chapters are of prime importance in the operation and design of furnaces.

## THE SELECTION OF FURNACES

In selecting a furnace the purpose for which it is required and the probable fuel consumption are points of primary importance. Its capacity should be closely related to the probable output since one of the principal sources of heat loss lies in incompletely filled furnaces. It is obviously a great deal more wasteful both in heat and labour to use two furnaces each loaded to less than full capacity to do work for which one would be adequate if operated at maximum capacity.

The cost of the fuel is a continuous charge on the heating operation, usually far outweighing the capital charges. Consequently the use of mechanical aids to operation, control instruments, and means of heat recovery should not be excluded on grounds of capital cost. An apparent high capital cost taken over the life of the furnace may be a considerable economy in the long run.

Other matters for special attention are :—

- (1) Robustness of the furnace structure. Besides a mechanically stable framework, good quality refractories and attention to efficient laying of the bricks are essential.
- (2) Ease of control. All valves, damper controls and instruments should be situated preferably in a convenient and, as far as possible, central position to give ready accessibility for control purposes. Automatic control, provided that over-elaboration is avoided, is an instrument of efficiency.
- (3) Ease of maintenance. This implies the use of readily replaceable parts, particularly those which are likely to be affected by heat or wear, and the means for inspection and cleaning.
- (4) Insulation, dealt with later, is to be regarded as an essential feature of almost any type of furnace.

- (5) Provision for adequate draught. A chimney is not necessarily the most efficient means of providing draught. Fans should be carefully rated for the duty intended. Flues, dampers and sight holes should be so arranged and constructed as to reduce to a minimum the possibility of air leakage.
- (6) Furnace atmosphere. It may be necessary to heat the goods in the furnace in an atmosphere having a composition that will not injure them.
- (7) Recovery of waste heat.

## FUEL SELECTION

Fuel selection, which is a highly important matter in furnace management, is discussed in Chapter XXXIV.

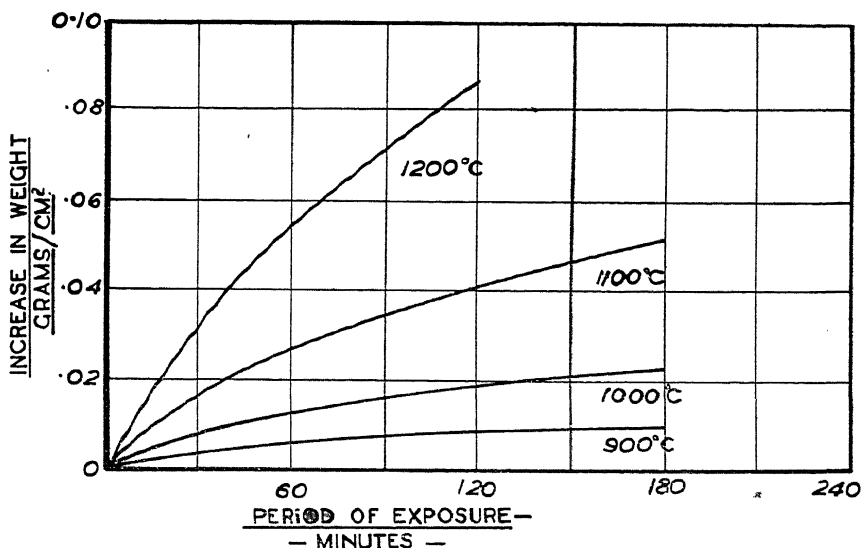


FIG. 129. Rate of scaling of mild steel in a neutral atmosphere.

(H. C. Millett and J. W. Cobb, *Trans. Inst. Gas Eng.*, 85, 672.)

## FURNACE ATMOSPHERES

The importance of the control of the atmosphere in a furnace has been widely recognised and this may be the dominant consideration in the selection of a fuel.

(a) *Internally Heated Furnaces.* The scaling of metals, particularly the readily oxidised non-ferrous metals, is of particular importance in furnaces in which the products of combustion of the fuel come in contact with the goods in the furnace. Similarly in the steel industry a reduction in scaling during heating could effect important economies. Thus the loss of 1 per cent. by scaling on an annual output of 10,000,000 tons would mean a direct loss of 100,000 tons of steel, valued at over £1,500,000. Steels are heated more than once in manufacturing processes and in addition to the direct scaling losses there are the indirect losses due to additional machining, wasters and roughened surfaces which affect coating operations. Not only does this mean a heavy wastage of steel, but also of the fuel used in making it.

Scaling has long been known to be due to the presence of oxidising gases and is in fact oxidation of the steel which becomes appreciable at temperatures above 700° C. The presence of free oxygen is not essential to scaling because



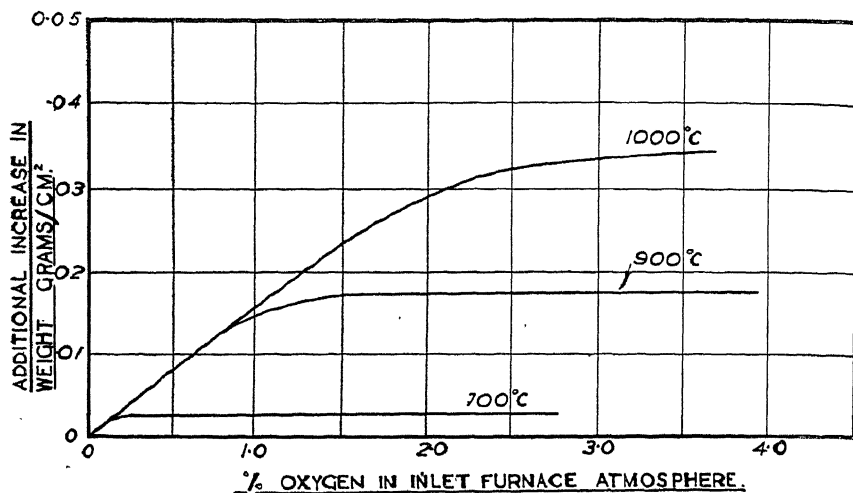


FIG. 130. Additional scaling of mild steel due to presence of free oxygen.  
(H. C. Millett and J. W. Cobb, *Trans. Inst. Gas Eng.*, 85, 675.)

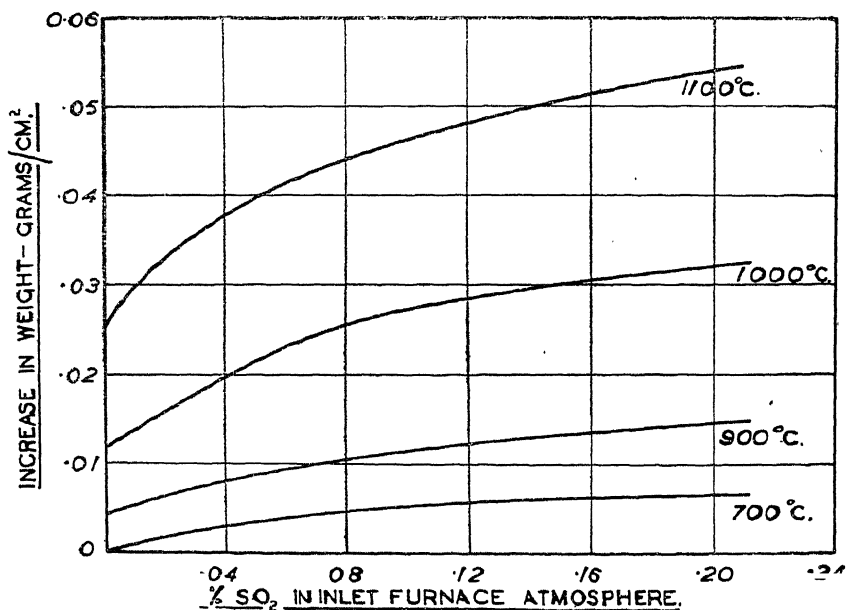


FIG. 131. Additional scaling of mild steel due to presence of free sulphur dioxide.  
(H. C. Millett and J. W. Cobb, *Trans. Inst. Gas Eng.*, 85, 671.)

carbon dioxide and water vapour may both give up their oxygen to iron. Fig. 129 shows the result of scaling experiments at Leeds University (*Trans. Inst. Gas E.*, 85, 672) indicating the effect of time and temperature in a "neutral" atmosphere, consisting of 10 per cent.  $\text{CO}_2$ , 20 per cent.  $\text{H}_2\text{O}$  and 70 per cent.  $\text{N}_2$ . The effect of a small percentage of oxygen to this atmosphere is given in Fig. 130.

If it is desired to inhibit scaling completely it is necessary to add hydrogen, carbon monoxide or methane in such large quantities as to be impracticable. The exclusion of oxygen to the extent that the products of combustion contain comparatively small amounts of carbon monoxide and hydrogen is sufficient so to reduce the oxidising effect as to minimise considerably the scaling of the steel.

The presence of oxygen or oxygenated bodies is not the only important condition for scaling. It has been found from the research at Leeds University just cited, that sulphur dioxide is a powerful scaling agent (cf. Fig. 131). The quantities of sulphur dioxide present in a furnace atmosphere are as follows:—

	$\text{SO}_2$ in furnace atmosphere per cent.
Purified coal gas (30 grains S per cubic foot) . . .	0.01
Coke with 1 per cent. S burnt with no excess air . .	0.07
Fuel oil, S = 1.5 per cent. burnt with no excess air	0.09
Coke with 2 per cent. S burnt with no excess air . .	0.14
Coal, S = 1.5 per cent. burnt with no excess air . .	0.15
Unpurified coal gas (600–700 grains S per cubic foot)	0.20

It is sometimes necessary to ensure that the atmosphere in a furnace shall be oxidising and sometimes that it shall be neutral or reducing to a definite extent. If an oxidising atmosphere is desired it is evidently necessary to use as little excess air as will complete combustion and give the desired atmosphere (cf. Chapter IV). When adjusting an appliance heated by coal gas to oxidising conditions, if the appliance is governed it is usual to adjust to 10.5\* per cent.  $\text{CO}_2$  and 1.5 per cent.  $\text{O}_2$ ; if the appliance is not governed a wider margin of safety is required and 9 per cent.  $\text{CO}_2$  should not be exceeded. The advantage in lower flue loss of the governed over the ungoverned appliance will be apparent and this is an indication of the desirability of expending money on control.

When the furnace atmosphere is to be adjusted to reducing conditions, as for steel hardening or similar processes, the primary and secondary air controls are regulated in conjunction with the flue damper to give a faint curtain of flame as the furnace door is opened. A piece of wood or paper thrown into the furnace should char, but not flame. This practical test indicates the absence of any appreciable amount of oxygen but gives no indication of the amount of CO present. Gas analysis should be used for this purpose and should show some 1–2 per cent. of CO present according to the process. Each 1 per cent. of CO produced by incomplete combustion is accompanied by 1–2 per cent. of hydrogen so that the potential heat loss in the unburnt gas is likely to be of the order of 7 per cent. for each 1 per cent. of carbon monoxide contained in the flue gas.

So-called reducing atmospheres can therefore be obtained by so adjusting the quantity of secondary air admitted to the furnace that there is produced a small quantity of carbon monoxide in the furnace gases, the amount that is present being determined by a simple form of gas analysis apparatus. Particular attention is directed to this fact because experience shows that many furnace operators seem to regard it necessary to pass large volumes of uncon-

\* This figure will vary somewhat according to the composition of the coal gas.

sumed gas through the furnace under the impression that only so will the atmosphere be reducing.

It has been established that carbonaceous particles such as smoke, exert little or no reducing action in a furnace. In order that any substance shall exert an effective reducing action it is clear that it must be brought into contact with the oxidisable materials in the furnace. The molecules of a gas are in a

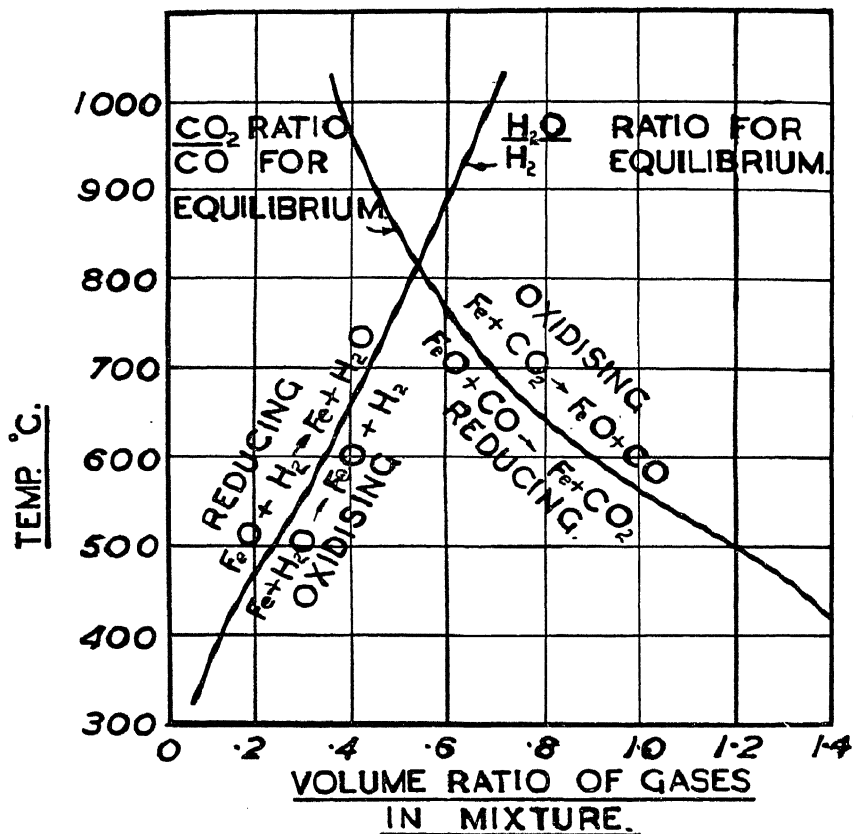


FIG. 132. Equilibria curves for reactions between steel and gases arising from combustion. The reactions printed on the left-hand side of the curves take place under the conditions existing on that side of the curves until the gas composition becomes that given by the curves; similarly for the reactions printed on the right-hand side, until the equilibrium composition of the gases in contact with the steel given on the curves is attained. Thus for a reducing atmosphere at any given temperature the gas composition must be maintained below the CO<sub>2</sub>/CO or H<sub>2</sub>O/H<sub>2</sub> ratios given by the respective curves for that temperature.

(Data due to Marshall and to Jominy and Murphy—reproduced from *G.E.C. Journal*, XII, No. 1.)

state of very rapid motion (the molecule of hydrogen, for example, at atmospheric temperature travels at the rate of 1 mile per second) and thus will come into contact with adjacent surfaces more freely and more frequently than particles of carbon which are conveyed through the furnace by the slow and comparatively gentle convection currents. Contrasted with the motion of gases smoke particles are relatively almost stationary.

(b) *Externally Heated Furnaces.* The bright annealing of metals is another process for which a carefully controlled reducing atmosphere is peculiarly essential. The primary object of most heat treatment operations is the attainment of a metallic structure having certain desired physical properties; the deleterious effect of the furnace atmospheres on the surface of the metal being

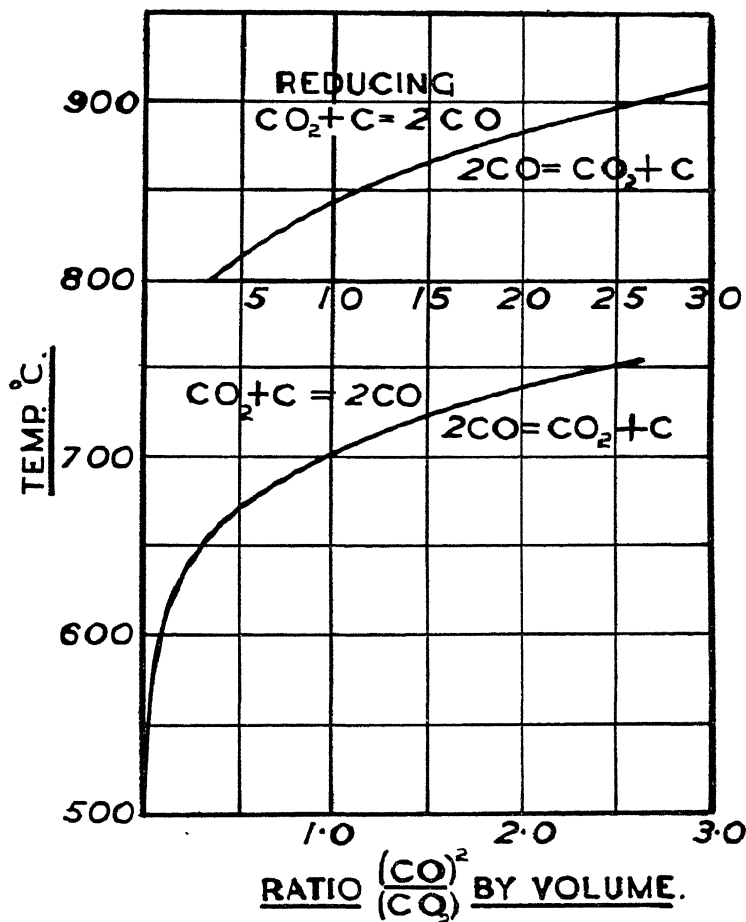


FIG. 133. Equilibria curves for the producer gas reaction,  $\text{CO}_2 + \text{C} = 2\text{CO}$ . Reaction  $\text{CO}_2 + \text{C} = 2\text{CO}$  takes place under conditions given on the left-hand side of the curves, and  $2\text{CO} = \text{CO}_2 + \text{C}$  under conditions on the right-hand side, until the gas composition at any temperature is that given by the curve.

(Upper curve due to T. F. E. Rhead and R. V. Wheeler and the lower curve to Marshall—reproduced from *G.E.C. Journal*, XII, No. 1.)

treated has been decreased for the relatively cruder operations of internally heated furnaces in the manner just described. One of the objectives of bright annealing is to eliminate scaling, decarburisation and other chemical reactions usually due to sulphur or oxygen which impair the surface of the metal. The problems of bright annealing vary with each metal used and the composition of the furnace atmosphere may have to be different for each metal.

There are very few metals or alloys which cannot be bright annealed, but with some the problem is more difficult than with others. For stainless steels with a high percentage of chromium the atmosphere must be completely free from oxygen and from oxygen-containing gases. The immunity from attack of this class of metal depends on an invisible film of oxide which would be broken up by further oxidation. For the bright annealing of steel, too, oxygen must be absent and the ratio of oxidising gases to reducing gases must be outside certain critical values. This problem demands a greater refinement in arrange-

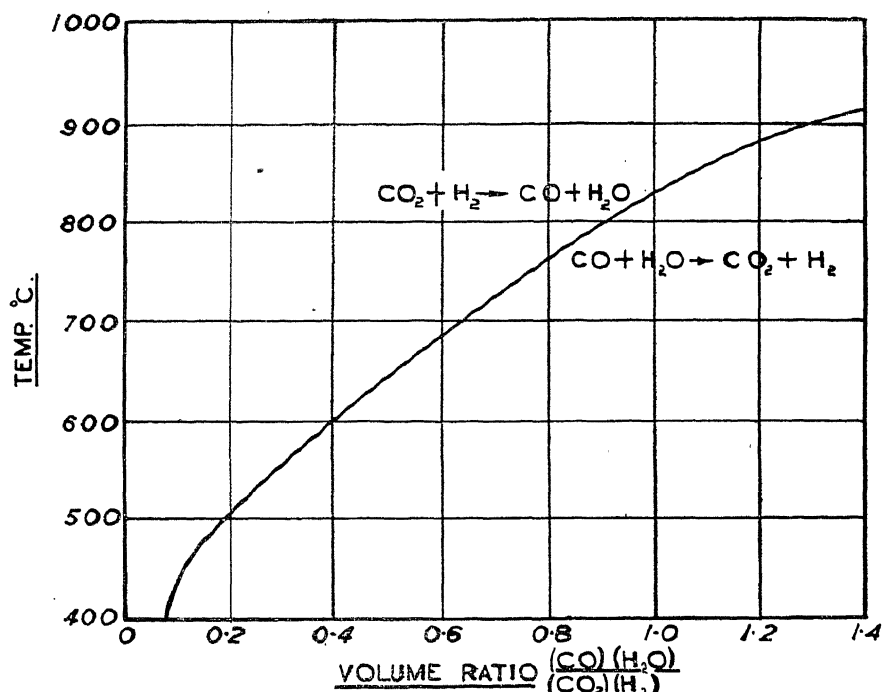


FIG. 134. Equilibrium curve for the water gas reaction,  $CO_2 + H_2 = CO + H_2O$ . Reaction  $CO_2 + H_2 = CO + H_2O$  takes place under conditions on the left-hand side of the curve, and  $CO + H_2O = CO_2 + H_2$  under conditions on the right-hand side, until the gas composition at any temperature is that given by the curve.

(Data due to Marshall—reproduced from *G.E.C. Journal*, XII, No. 1.)

ments for producing a furnace atmosphere than the scaling of ordinary mild steel previously discussed.

The necessary exact control of the atmosphere cannot be attained if the combustion gases circulate in the furnace. The furnace must be heated indirectly and the gases that are to constitute the furnace atmosphere must be produced separately in an ancillary apparatus.

Space does not permit of a detailed discussion of this problem here, but the oxidation of bright steel is preventable by keeping the ratios:  $CO_2/CO$ ;  $H_2O/H_2$ ;  $(CO)(H_2O)/(CO_2)(H_2)$  and  $(CO)^2/(CO_2)$  above or below certain limits, the values of which depend on the temperature, see Figs. 132, 133 and 134. A detailed discussion of this subject is to be found in the literature.

In general there is now no difficulty in partially burning gas so that the

resultant atmosphere satisfies the required conditions of composition. It is evident that the most careful control must be exercised over the combustion of gases in a furnace when the products can be affected by the furnace atmosphere.

(c) *Burners for Production of Furnace Atmosphere.* Special burners have been patented for this purpose in which gas and air are fed into the system by power-driven blowers with vernier proportioning control. These mixers govern both quantity and quality of the gas, and will control the  $\text{CO}_2$  content for example, to within 0.5 per cent. In this type of equipment, the unit may be pre-calibrated by analysis of the purging gas so that the correct conditions for bright annealing a variety of materials can be secured without trouble in practice by a simple setting with a pointer.

Where water vapour must be eliminated, it is absorbed in silica gel. The gases are burnt in a refractory-lined combustion chamber and give up part of their heat by passing in counter-current to the air used for drying the silica gel chambers. The gases are then passed through a condenser and cooled to atmospheric temperature to remove the bulk of the water arising from the combustion of hydrogen. They are then dried by silica gel in a special chamber, and are passed through oxide of iron boxes for removal of sulphur. Silica gel absorbs the last traces of water and is revived by passing hot air over it. The furnace is, of course, kept under a slight pressure to prevent ingress of air. There are many alternative methods of producing the necessary atmosphere, as for example by the decomposition of ammonia.

## COMBUSTION AND HEAT TRANSMISSION

In the operation of any equipment where fuel is burnt the correct proportioning of the fuel and air, and also of the primary and secondary air, are necessary for two reasons :—

- (1) To ensure optimum combustion conditions so that the quantity of fuel used shall be the minimum.
- (2) To produce continually and with certainty the correct furnace atmosphere.

The type of fuel used and the method of burning it will be largely controlled by the degree of importance of these two conditions.

(a) *Control of Combustion.* The combustion of coal has been dealt with extensively in earlier chapters dealing with the boiler furnace. From what has been said there it will be appreciated that it is impossible to control the furnace atmosphere by hand firing solid fuel and that hand firing is only to be used in furnaces where fine control of temperature or atmosphere is not required.

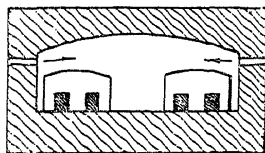
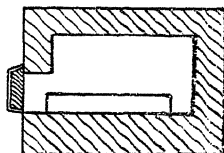
With mechanical stokers, and more particularly with powdered fuel firing, greater control can be secured over both furnace atmosphere and temperature because of the more uniform admission of fuel and air. Solid fuel is less suitable than gas if controlled fluctuations in temperature are used, or if exact temperature control is required. The use of hot crude producer gas is susceptible of greater control than solid fuel and is approached by powdered fuel firing in this respect. For certain operations powdered fuel firing has the disadvantage of introducing all the ash from the coal into the furnace whereas in crude producer gas the content of inorganic dust is very much less.

For exact control of combustion conditions it is necessary to use cold, cleaned producer gas or coal gas both of which can be valve controlled and can thus be used to produce not only the required temperature but also the required furnace atmosphere.

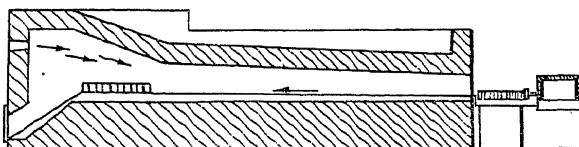
(b) *Control of Flame.* The general principles of combustion have already been described in earlier chapters and no further reference to them need be made here. In furnaces it is necessary to control the temperature, and the method

by which the goods are heated. Furnaces can be subdivided into two general types:—

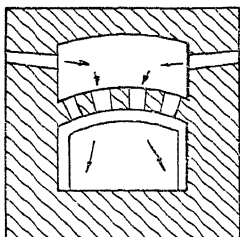
- (1) Furnaces indirectly heated (e.g. muffle furnaces) in which the heat is generated in flues or in a separate combustion chamber and must pass through a refractory or other material forming the furnace wall into the



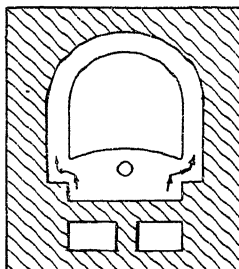
BATCH FURNACE.



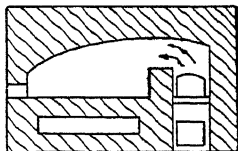
CONTINUOUS FURNACE WITH END DISCHARGE.



OVER-FIRED FURNACE.



MUFFLE FURNACE.



COAL-FIRED FURNACE.

**FIG. 135.** Types of furnaces.

Reproduced by permission of the publishers (John Wiley and Sons) from "Industrial Furnaces," by W. Trinks.)

heating chamber proper in which the goods are placed. The products of combustion do not come in contact with the goods.

- (2) Directly heated furnaces in which the flame may be developed within the chamber in which the stock is placed or in a separate furnace, but in either event the products of combustion come into contact with the goods.

In the first type of furnace the principles of combustion and heat trans-

mission which have previously been established in earlier chapters hold without modification. The stock is to be heated to a certain temperature and according to this temperature the temperature in the flues is regulated by the total quantity of fuel burnt. The objective must evidently be to burn the fuel with the maximum efficiency and to pass the maximum quantity of the heat generated through the furnace wall. The heat is generally transmitted to the stock partly by convection and partly by radiation from the interior surface of the furnace walls.

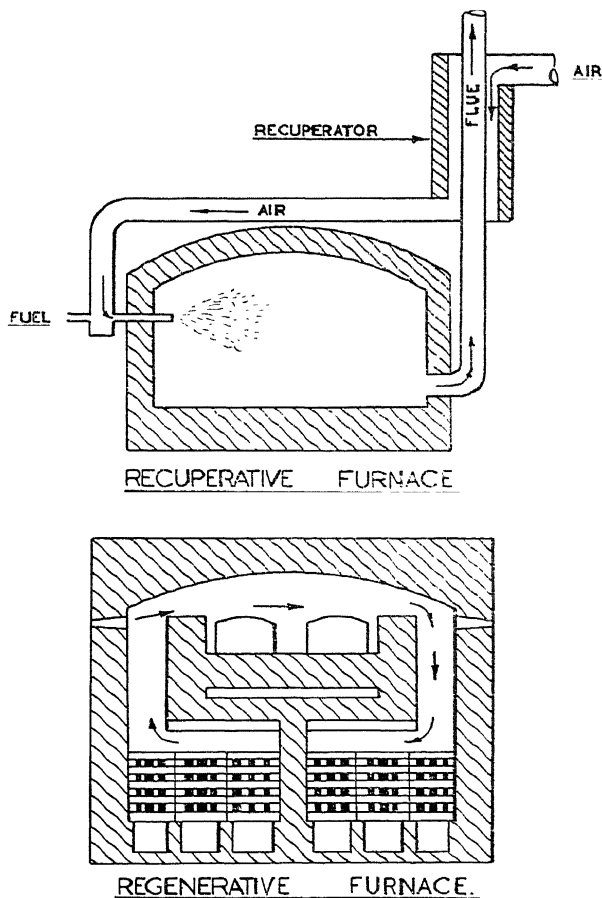


Fig. 135. Types of furnaces.

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The directly heated furnace involves all methods of heat transfer—radiation from the roof and sides of the furnace; convection and conduction from the hot gases flowing through the furnace; and gas radiation from  $\text{CO}_2$  and  $\text{H}_2\text{O}$  in the heating gases. A number of furnaces are sketched diagrammatically in Fig. 135 (pp. 384/5) which will illustrate these general types.

Attention has been previously directed to the fact that flames can be chilled and caused to deposit carbon by contact with cold surfaces. If, therefore, a



flame from the volatile products of coal or from coal gas is allowed to impinge upon cold stock in the furnace, smoke will be produced and soot will be deposited on the stock. Any combustible, even the constituents of gaseous fuels or oil, can be chilled in the same manner. Moreover, heating by flame impact would frequently be too fierce and would spoil the material heated. The flame must generally therefore be so controlled and directed that it does not impinge directly upon the stock.

If the heat is to be transmitted primarily by convection and gas radiation aided by gentle radiation from the furnace brickwork a short high-temperature flame is developed within the combustion space and the hot gases are then allowed to stream into the furnace chamber and come in contact with the stock. If, on the other hand, the greatest possible assistance is to be derived from radiation a long lazy flame is desired which will be at a lower temperature, but which will continue for a long distance into the furnace and so will heat the furnace roof and radiate directly on to the charge.

The short high-temperature flame is developed by intimate mixing of the gas and air, preferably with the air and gas (if it be low grade gas) preheated and with as turbulent a flow as possible.

The long lazy flame is obtained by admitting air and gas in parallel streams through separate ports and allowing them to mix slowly by diffusion as they pass through the furnace. Flames of this type may be as much as 30 feet in length. Thus the continuous furnace with end discharge shown in Fig. 135 may have a long flame starting at the left-hand side and extending a considerable distance into the furnace.

(c) *Depth of Flame.* The flame is frequently an index of whether the hot gases are flowing in such a manner that the hot products of combustion are coming in contact with the stock, and so adding the forced convection effect to that of radiation. Accordingly the dimensions of the cross-section of the furnace require to be arranged in keeping with the volume of products of combustion, whereby the flow does not become stratified in an ineffective manner such as to keep it out of contact with the charge. The theoretical works of Groume-Grjmaillo and Yesmann ("The Flow of Gases in Furnaces" by Groume-Grjmaillo) should be consulted on this point.

(d) *Mode of Heat Transfer.* As an example of the method of heat transfer, it has been shown that in a rolling mill furnace where the maximum temperatures are of the order of 1,100°–1,300° C., the distribution of the modes of heat transfer is as in Table 80.

TABLE 80

Method of heat transfer	Percentage of total heat transfer
Direct gas radiation .. .. .	54
Furnace surface radiation .. .. .	34
Convection .. .. .	12

(e) *Size of Combustion Chamber.* The rate at which fuel can be burnt determines the size of the combustion chamber. The condition of the fuel and the temperature of preheat together with the method of mixing have an important bearing upon the dimensions. Some good practical rules are as follows:—

- (1) Poor mixing of fuel with cold air and poor utilisation of combustion space :

Approximately 1.5 B.Th.U. per cubic foot per second.

- (2) Reasonably good mixing of fuel with preheated air and fair utilisation of combustion space :  
Approximately 3-4 B.Th.U. per cubic foot per second.
- (3) Good mixing of fuel and preheated air and good utilisation of combustion space :  
Approximately 6-10 B.Th.U. per cubic foot per second.
- (4) Very fine atomisation of fuel and perfect mixing with highly preheated air, with excellent use of combustion space :  
Approximately 10-20 B.Th.U. per cubic foot per second.

Experience has shown that for annealing and reheating furnaces generally, condition (2) is usually attained, whilst for high temperatures such as melting furnaces condition (4) is not unusual. Much higher intensities are practicable, for example, 70-190 B.Th.U. per cubic foot per second, with the type of heating known as surface combustion, in which the gases are made to unite in contact with a refractory surface, accelerating combustion by surface catalysis. A burner firing into a hemispherical refractory cavity has been developed in the United States of America, for which far higher rates of heat release are claimed. The problem of the location of burner and combustion space in these examples is resolved into one of the mode of heat transfer in the furnace itself.

(f) *Draught and Gas Velocities.* Conjointly with the estimation of the heat release from the flame, there must be considered the question as to whether the gases can be evacuated from the furnace chamber without undue loss of draught. An upper limit of 30 feet per second for velocity of the hot gases is usually required for this reason except in large melting furnaces, such as steel and glass-making furnaces, where higher gas velocities may be necessary to give quick melting.

For practical design the rates of flow are important. The following velocities are useful in calculating flue and valve sizes :—

Cold air under natural draught, 5-10 feet per second. Air or gas for temperatures up to about 500° C. (940° F.) 10-15 feet per second.

Products of combustion from 750° C. (1,380° F.) to 1,250° C. (2,280° F.) 15-25 feet per second.

When calculating sizes of flues and ports and laying out the design, care must be taken to ensure that there are no sudden restrictions where the velocity has to be increased considerably, and all bends and corners should be eased as much as possible. This subject was discussed in detail in Chapter IX, where a specimen calculation was given.

(g) *Rate of Heating-up Stock.* The rate at which heat is transmitted to materials depends on their diffusivity. As will have been seen from Table 36 (Chapter VIII) the thermal conductivity of metals is high, but so is the specific

TABLE 81

Metal	Conductivity, B.Th.U./sq. ft./ ° F./hr./ft. thickness $k$	Specific heat, B.Th.U./lb.° F. over range 0-600° F. $s$	Density lb./cu. ft. $\rho$	Diffusivity sq. ft./hr. $\frac{k}{\rho s}$
Copper ..	220 (200-1,000° F.)	0.11	558	3.58
Aluminium ..	108 (at 600° F.)	0.24	168	2.68
Brass ..	66 (at 600° F.)	0.092	530	1.37
W.I. ..	35 (at 200° F.)	0.12	458	0.64
C.I. ..	26 (at 200° F.)	0.13	442	0.44
Mild steel ..	25 (at 200° F.)	0.115	488	0.45
Firebrick ..	0.6 (at 600° F.)	0.23	125	0.024

gravity. The value of the diffusivity compared with that of refractory materials is high. Thus, expressing the diffusivity as the rise in temperature produced in 1 cubic foot of the material by 1 B.Th.U. per hour, the diffusivities of several metals are as in Table 81.

Approximate practical rates of heating have been given for iron and steel (Webb, *Trans. Inst. Chem. E.*, 1938), as follows:—

Heat treating	..	..	..	..	30	lb./sq. ft./hr.
Annealing	..	..	..	..	60	" "
Forging	..	..	..	..	80	" "
In continuous furnaces	...	..	..	..	50-100	" "

All the foregoing figures depend on the thickness of the stock.

The figures refer to a hearth of which the area is completely covered by the material being heated. For large sized pieces useful figures for low carbon steels are 16-20 minutes per inch thickness and for high carbon steels 30-35 minutes per inch thickness.

As an example of the use of these figures (quoted by C. Webb, loc. cit.), let it be assumed that bars 2 inches square, 6 feet long, are to be annealed and an output of 30 bars per hour is required. The weight of the bars is 81.6 lb. each; therefore the output is 2,448 lb. per hour. Taking a heating rate of 60 lb. per square foot per hour the area required is 40.8 square feet. As the bars are 6 feet long, a hearth width of 6.8 feet will be required.

For a continuous furnace, for 2 inch square bars, 6 feet long, to give an output of 100 bars per hour, the heating rate with a well-designed furnace would be about 90 lb. per square foot per hour: the length of the furnace would be 15.2 feet.

As an example of heating a large round ingot, say 20 inches diameter by 6 feet long, the maximum figure above should be taken, namely, 20 minutes per inch, giving 400 minutes for heating. This would apply if the ingot were heated on its side. If the ingot were placed vertically, as in a soaking pit, so that the heat could penetrate from all sides, the heating rate could be taken as 16 minutes per inch, reducing the time to 320 minutes.

The rate at which non-ferrous metals can be heated compared with iron and steel is proportional to the following fraction:—

$$\frac{\text{Conductivity of steel} \times \text{specific heat of non-ferrous metal}}{\text{Conductivity of other metal} \times \text{specific heat of steel.}}$$

To obtain the size of furnace required, divide the lb. per square foot per hour for steel by the value of the fraction. This rule does not apply to metals of high conductivity such as copper and aluminium, which can be heated at rates giving approximately 55 per cent. to 65 per cent. and 30 per cent. to 35 per cent. respectively of the times for heating steel.

Although the rate of heating depends upon the diffusivity as mentioned, it must be confessed that heating rates are in the present state of knowledge based on empirical rules, arrived at as the result of practical experience; as examples there are the "practical" rules (1) that heat penetrates ordinary carbon steel at the rate of  $\frac{1}{8}$  inch in five minutes; (2) that highly alloyed steels, which have generally a lower thermal conductivity, require at least one hour per inch, and that whilst heating must be slow up to 500°, it can be speeded up beyond this point. Another basis is to relate the times of heating to the output on a given hearth area; thus practical figures quoted by the steel industry are 30 lb. per square foot per hour for heat treating, 40 lb. for general rolling and forging of large blooms, and 60-80 lb. for carbon steel billets below 4 inches in linear section. These rules are not altogether in agreement with the figures quoted

from C. Webb's paper. Some further discussion on this subject will be found at the end of Chapter XIX.

Recent investigation has shown that with metals more depends upon the manner in which the heat is distributed to heat the furnace walls and bring them to temperature uniformity than upon the thermal constants or dimensions of the charge.

The practical points of control in arriving at efficient working are as follows :—

- (1) The effect of different methods of disposing the charge should be tried until that which gives the maximum output per unit time is obtained.
- (2) The mass of the charge should be varied, if the operation admits of such a procedure, until a point at which an increase of weight per unit area of hearth either disturbs the final desired distribution of temperature at the finishing point or the rate of output becomes reduced. Generally the limiting factor is determined by the nature of the operation, and undercharging is more common than overcharging. The ideal hearth area for a specific duty should be sought. Much fuel can be wasted in wrong-sized furnaces.
- (3) The rate of firing should be changed experimentally until the optimum conditions are found, and then valves and dampers should be marked to show the ideal settings.

## SOURCES OF HEAT LOSS IN FURNACES

Fig. 136 derived from Trinks' "Industrial Furnaces," summarises the movement of heat in a furnace. The heat is generated in the combustion chamber

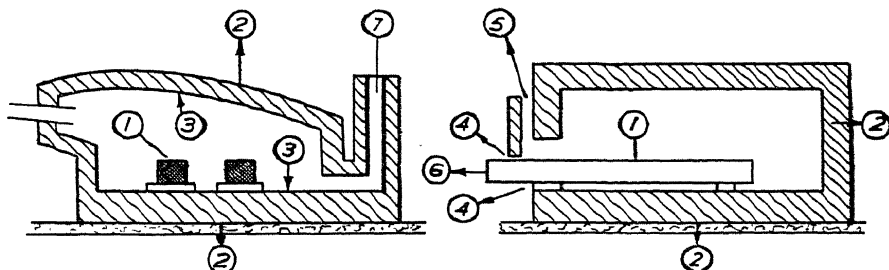


FIG. 136. Heat transmission and heat losses in furnaces.

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at the left, and travels from there to the right. The passage of heat into the stock, as indicated by arrows (1), is desired. But heat also goes elsewhere; some of it passes into the furnace walls and some into the hearth, increasing their temperature as indicated by the arrows (3). Another portion of the heat is lost to the surroundings by radiation and convection from the outer surface of the walls, or by conduction into the ground (2). Through cracks or other openings, heat radiates away (4); and furnace gases pass out around the door (5), frequently burning in the open, and carrying off heat. Then there are the special losses, peculiar to certain types of furnace: for example, in furnaces which heat only part of a long piece of metal, heat passes out along the metal from the part which is in the furnace to the part which is outside, and is there dissipated to the surroundings (6). Radiation from the exposed liquid surface of salt baths or lead baths, and conduction of heat through the terminals or electrodes of electric furnaces, are other cases in point. Water cooling of skid pipes and of conveyor rollers absorbs large quantities of heat and lowers thermal efficiency. If the charge is heated in containers or on travelling

chains, a large part of the heat is dissipated in the open by these devices, after they have been brought up to furnace heat. Finally, heat passes out with the products of combustion (7), either in the form of sensible heat or as undeveloped heat of combustibles escaping unburned (incomplete combustion).

### THERMAL EFFICIENCY

The term thermal efficiency as used for boilers is not necessarily applicable to furnaces because of the different purposes for which heat is used. In a boiler, heat is applied to water for the purpose of converting it into steam which is used in that condition. In a furnace, the heat is given to a substance not for the purpose of raising its temperature *per se*, but in order that certain chemical or physical changes shall take place in the material.

In the burning of bricks for example it is not only necessary for the material to be raised in temperature to, say, 1,500° C., but a certain time must be taken over each of the stages in order that (1) the shapes may be dried without cracking, (2) carbonaceous matter contained in them may be oxidised, and (3) that clay may be converted into mullite, or (with silica bricks) that quartz may be converted into tridymite and cristobalite. The process requires that the temperatures shall be attained slowly and shall be held for a period which may be measured in days. Thus the term "thermal efficiency" as defined in a straight heat transmission process is not necessarily applicable to furnaces.

In the carburising of mild steel the useful heat employed in raising the steel to carburising temperature is only a part of the process; during by far the greater period when the steel is in the furnace there is no heat flow to the steel, since the diffusion of carbon in steel is a reaction requiring time at a particular temperature for its completion.

Another example is the salt bath treatment in the normalising of certain aluminium alloys. But little heat is required to raise the metal to its normalising temperature, 495° C., and the transfer takes place very rapidly; thenceforth the process requires that a temperature of 495° C. shall be maintained in order that the necessary changes can take place in the metal. The heat required by the charge during this second stage is very small indeed, and fuel is required essentially to make up for the heat loss from the salt bath.

In these examples the efficiency of the appliance bears no relation to the real efficiency of the process.

For boilers, the thermal efficiency can be expressed as:—

$$\begin{aligned} \text{Boiler efficiency} &= \frac{\text{heat in steam}}{\text{total heat input}} \\ &= \frac{\text{Total heat input} - \text{flue loss} - \text{surface loss}}{\text{total heat input.}} \end{aligned}$$

But, the efficiency of a furnace must usually be related to the real work done as expressed in the term:—

$$\text{Real efficiency} = \frac{\text{total fuel consumption}}{\text{quantity of work heat treated.}}$$

Thus the expression thermal efficiency as a general criterion for furnace operations is meaningless and each operation must be considered on its merits. It is for this reason that thermal efficiencies in furnace operation are often very low, not because of inefficiency, but because of the characteristics of the process.

This, however, does not mean that there is not a great deal of inefficient operation in furnaces and the low heat efficiencies often obtained can be very materially improved by making use of the principles of fuel technology.

## HEAT BALANCES

As examples of the heat balances of furnaces and as indicating the distribution of the heat, there are reproduced in Table 82 details of heat losses from typical furnaces.

TABLE 82. DISTRIBUTION OF HEAT LOSSES IN TYPICAL FURNACES

Type of furnace	Ceramic kiln	Producer gas fired recuperative billet heating	Recuperative coal fired forging	Non-continuous regenerative gas-fired heating	Oil-fired forge	Coke-fired forge	Coal-fired annealing	Recuperative gas-fired reheating
Source of loss..								
1. Useful heat transferred to charge ..	14	23	16.1	40.9	9	16.1	18	48.7
2. Combustion and chimney losses								
(a) Unburnt combustible ..	3	2.6	3.0	2	3	5	5	1.0
(b) Sensible and latent heat in waste gases..	50	22.4	47.6	26.7	47	52.3	45	3.9
(c) Gas generation loss in producer gas fired furnaces ..	—	15.0	—	—	—	—	—	—
3. Furnace structure in brickwork and heat lost to surroundings..	33	37	33.3	30.4	41	26.6	32	46.4

## UNAVOIDABLE HEAT LOSSES

As in boiler plant so also in furnaces there are certain heat losses which may be taken as unavoidable. Reference has been made to some of these in Fig. 136.

(a) There must inevitably be a certain loss of heat in the flue gas. After all industrially possible has been done to extract the useful heat from these gases there are certain losses which are inevitable. If a chimney is used for creating the draught the requisite temperature must be retained in the gases for operating the chimney. Even if a fan is used for operation it is still not possible to reduce the temperature of the gases below about 300° F. because there is no way of making use of low grade heat usefully. The latent and sensible heat of the water arising from combustion and (with wet material) evaporated from within the furnace must also be lost.

(b) However excellent the insulation of the furnace may be there must inevitably be a small escape of heat through the walls to the atmosphere.

(c) An important source of heat loss is that arising from the sensible heat of the products in the furnaces. In some continuous processes, as for example in continuous gas retorts and continuous brick kilns, it is possible to recover the heat in the products by using them to produce steam or by circulating air round them which is subsequently used for combustion. In a great many instances, however, as for example in rolling mills, the purpose of the heating operation is to raise the stock to a high temperature at which it is to be used for some succeeding process. In that event the heat taken away from the furnace in the sensible heat of the stock cannot be recovered.

A borderline example is that of the coke in by-product coke ovens. This is discharged at 1,000° C. and represents something like 40–45 per cent. of the heat put into the flues to effect the carbonisation, but generally it is lost when the coke is cooled with water; in some installations this heat is recovered by a

dry quenching plant in which inert gases are circulated through the hot coke until it is cooled and are then circulated through a waste heat boiler so that something like 900–1,000 lb. of steam can be raised per ton of hot coke. This, however, is not a pure problem in heat recovery but depends on the economics of the process.

(d) If gas producers are used in connection with the furnaces there must also be a certain thermal loss due to the operation of this section of the plant. A cold gas efficiency of some 60–75 per cent. or hot gas efficiency of 70–85 per cent. may be expected in efficient producer practice.

### AVOIDABLE HEAT LOSSES

There are several avoidable losses to which particular attention must be devoted. These will be the subject of succeeding sections in this chapter and may be summarised as follows:—

Excess heat in flue gas.

Excessive loss by radiation, etc., from outer furnace walls.

Loss by radiation through doors or other openings in the furnaces.

Losses by escape of furnace gases.

### RECOVERY OF HEAT FROM FLUE GASES—RECUPERATION AND REGENERATION

A furnace may be considered as consisting of a combustion chamber and a chamber in which the stock is heated. These may be separate or the same chamber may be used for both purposes.

In continuous furnaces the hot gases generally pass in counter-current to the cold incoming stock and there is thus opportunity, if the furnaces be long enough, for efficient heat transfer from one to the other, the gases leaving the furnaces at a temperature so low that there is only just sufficient heat to operate the chimney and no further recovery is possible or economic.

In intermittent furnaces, on the other hand, the outlet temperature of the gases must depend upon the temperature to which the stock is required to be heated. In the earlier stages of heating the outlet temperature is lower than in the later stages, the exact temperature depending on the conditions in the furnace, as for example on whether the furnace is maintained at a high temperature continuously or whether, as in brick kilns, it is allowed to cool down between each operation.

For the purpose of this discussion it is assumed that the combustion gases leave the furnace at a high temperature as in glass furnaces, open-hearth furnaces, gas retorts or coke ovens. Under these conditions some use must be made of the surplus heat in the waste gases.

It can be calculated as for example in Fig. 165, Chapter XX, that if the products of combustion with normal amounts of excess air escape at 1,000° C. they carry away from the furnace nearly 6,000 B.Th.U. per lb. of fuel used in

TABLE 83. SENSIBLE HEAT IN PRODUCTS OF COMBUSTION FROM VARIOUS FUELS. PER CENT. OF GROSS POTENTIAL HEAT

Temperature				1,200° C.	600° C.
Coal	14,500 B.Th.U./lb.	..	..	48.3	21.7
	12,750 B.Th.U./lb.	..	..	49.3	21.5
Fuel oil	..	..	..	46.8	21.5
Coke oven gas	..	..	..	48.1	20.9
Producer gas	..	..	..	65.6	29.8
Blast furnace gas	..	..	..	79.1	34.2

## FURNACES

the producer, or about 45–50 per cent. of heat initially supplied to the process (see also Chapter V). Table 83 gives further data on this subject, and refers to sensible heat only, omitting the latent heat of water vapour contained in the gases.

Whether this heat can be recovered or not depends in the first instance upon the scale of the operations. Obviously heat recovery cannot be applied to a small rivet-heating furnace treating a matter of pounds per hour, whereas it could be applied to an open-hearth furnace dealing with an output measured in tons per hour. Plant is available now to treat the gases from quite small furnaces.

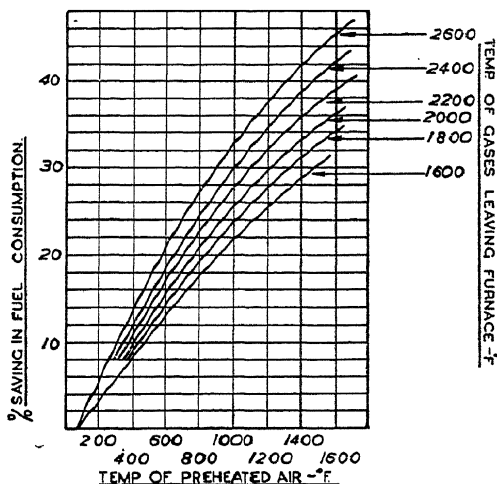


FIG. 137. Saving of fuel in powdered fuel furnaces gained by preheating the air for combustion.

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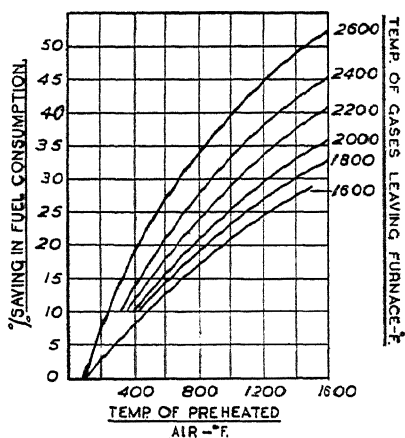


FIG. 138. Saving of fuel in producer gas fired furnaces gained by heating the air for combustion.

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Assuming that the scale of operations makes it economic, three methods of heat recovery from the flue gases are practised :—

- (1) Recuperation.
- (2) Regeneration.
- (3) Waste heat boilers.

Recuperation and regeneration are methods by which the air used for combustion, and sometimes the gas also, are preheated by making use of the sensible heat of the flue gases leaving the operative part of the setting. Whether these methods can be applied depends on whether the process is one for which preheated air is desirable. For most heating operations in which a high temperature is required preheat is desirable. It has been pointed out in Chapter V that by preheating the air and gas the flame temperature is very considerably increased. If heat is to be transferred by direct radiation from the flame or from the hot gases, the flame temperature should obviously be as high as possible and air and gas should be preheated. On the other hand, if a long, lazy, low temperature flame is required preheated air and gas are undesirable.

It is not possible to preheat coal gas because it decomposes at a dull red heat with deposition of carbon. It is equally impossible to preheat solid fuels



(though as an exception to this, red-hot coke straight from the retorts is used in producers on gas works). In general producer gas and blast furnace gas are subjected to preheat.

Air for combustion can always be preheated if it is desirable to do so.

If there is efficient heat interchange, for solid fuel firing and for coal gas firing the quantities of incoming air and outgoing flue gases are sufficiently similar for the process of preheating the air to reduce the temperature of the flue gases to some  $200^{\circ}$  or  $250^{\circ}$  C., below which they are not cooled if they are used for operating a chimney. For producer gas and blast furnace gas, however, there is a surplus of heat in the flue gases over that which could be given to the incoming air and consequently if the air only is preheated (as in gas retorts) the temperature of the flue gases leaving the heat interchange plant may still be high enough to warrant the installation of a waste heat boiler.

Waste heat boilers may thus be installed after regenerators (a) when air is being preheated for the combustion of producer gas or blast furnace gas, and (b) when only a moderate degree of preheating is permissible. The saving of heat effected by preheating air is illustrated in Figs. 137 and 138.

Waste heat boilers are dealt with in Chapter XX and will not be further referred to here.

Recuperators consist broadly of systems of flues, some of which carry ingoing air and others outgoing flue gases at a higher temperature, so arranged that there can be heat interchange between the air and gas.

Recuperators are of three types, counter-flow (Fig. 139), parallel-flow (Fig. 140) and cross-flow (Fig. 141). These figures also show the general form of the temperature curves.

The counter-flow type allows the highest temperature of preheat to be reached, whereas the parallel-flow type gives the lowest maximum temperature of the recuperator walls. It is frequently used in conjunction with metallic recuperator walls where the temperature must be kept comparatively low.

Recuperators depend on heat transferred by conduction through a wall of material between two streams of gas, and may be less efficient thermally than regenerators especially when constructed in brick. They have, however, the advantage that they do not require periodic reversal because they are fixed passages through which air and gas flow continuously.

A regenerative furnace has been sketched diagrammatically in Figs. 49 and 135.

A regenerator is a heat exchanger constructed of refractory material. Reference to Fig. 135 will show that in its simplest form it comprises two chambers filled with chequer firebricks, the bricks being so stacked that the gases can flow freely between them and around them. If gas is burned as shown in Fig. 135 from the port on the left-hand side, the hot flue gases after doing their work in the furnace pass down the right-hand regenerator chamber and give up much of their sensible heat to the brickwork. The temperature of the brickwork decreases from top to bottom of the chamber. The air for combustion flows up the left-hand chamber. After some predetermined period, usually from 20 minutes to 30 minutes, a reversing mechanism changes the direction of flow of the gases. The gas is cut off from the left-hand ports and now admitted to the right-hand ports. The right-hand regenerator is disconnected from the chimney flue by closing a valve. The air is admitted to the base of the right-hand regenerator, and the flue gases pass down the left-hand regenerator. Thus the right-hand regenerator now gives up its stored heat to the incoming air, which is preheated, and the sensible heat of the gases is retained in the left-hand regenerator. After 20–30 minutes the flow is again reversed and the cycle is repeated.

An extension of the system permits alternate regenerator chambers to be

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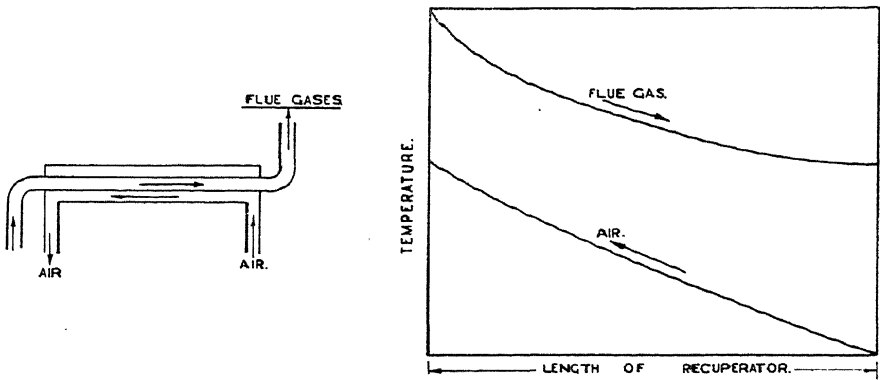


FIG. 139. Counter-flow recuperator and temperature distribution.  
(Based on "Industrial Furnaces" by W. Trinks.)

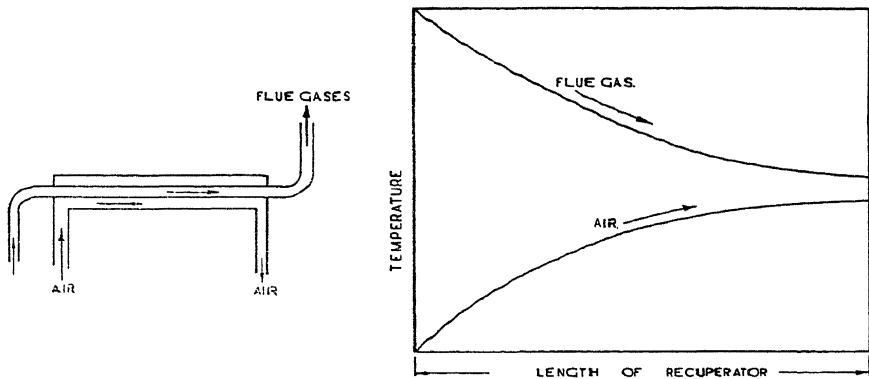


FIG. 140. Parallel-flow recuperator and temperature distribution.  
(Based on "Industrial Furnaces" by W. Trinks.)

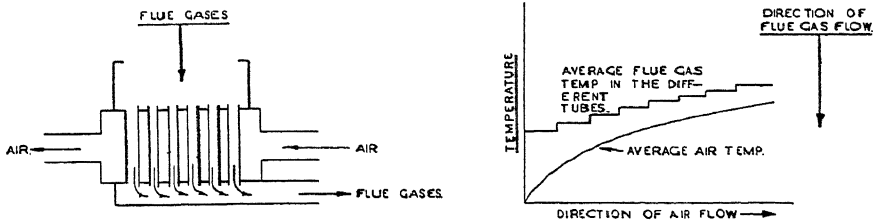


FIG. 141. Cross-flow recuperator and temperature distribution.  
(Based on "Industrial Furnaces" by W. Trinks.)

used for preheating producer gas or blast furnace gas, the air being also preheated.

Regenerator design is generally a compromise between the requirements of mechanical strength, freedom from choking due to dust or fume in the products of combustion, and heat transmission surface. Design has been dealt with in a number of publications (e.g. *Journ. Inst. Fuel*, 4 (1931), 160-174; *Journ. I.S.I. Special Report*, No. 22, 1938, pp. 238-275).

The temperature conditions are analogous to those which occur in a counter-flow recuperator. From the standpoint of control there is already means of improving the efficiency of regenerators at rebuilds, since the change only involves modification of methods of stacking the regenerator chequer brickwork. The subject is beyond the scope of the present discussion, but merits close investigation on the part of all users of regenerators.

To combine increased heat-absorption surface with an economical use of brickwork, special shapes are sometimes used, but it is still uncertain whether such expedients are necessary in regenerators of adequate design.

Between 60 and 85 per cent. of the sensible heat removed from the waste gases during passage through a regenerator is transferred to the air and/or gas being preheated. Taking, however, the heat returned to the furnace by the regenerators via gas and air, as a percentage of the sensible heat in the gases reaching them, which constitutes the true thermal efficiency of the regenerative system exclusive of the flues, the maximum value realisable is of the order of 40 per cent., depending on the degree of preheat which is admissible.

## INSULATION

The furnace wall has a threefold function : (1) to ensure the necessary refractoriness to withstand the effect of the flame and the slagging action of the dust in the furnace atmosphere ; (2) heat insulation ; (3) in certain processes, to provide a means of supplying heat to the stock by radiation.

It is not generally possible to obtain an ideal combination of these properties in any one material. Accordingly, a composite wall of firebrick and insulating brick is used. The effectiveness of the insulating materials depends upon the minuteness and uniform distribution of the cells of air present in the material, the heat conductivity of which is extremely low.

The physics of insulation has been considered in Chapter VIII and the properties of insulating materials will be discussed in Chapter XXII. Here, only the practical application of the subject will be considered. The temperature at which any insulating materials can be used is limited. When a refractory material forming a furnace lining is backed by insulation, the temperature of the outer surface of the refractory forming the interface between it and the insulation is raised, and so also is the average temperature of the brick. The interfacial temperature is the maximum temperature to which the insulation is heated and this must be less than the limiting temperature given in Chapter XXII. Furthermore, as indicated in that chapter, the effect of insulation may be to bring the refractory within the danger zone at which it can collapse under load. Losses of heat and interfacial temperatures are given for a number of constructions in Table 84. The insulation of furnaces is extremely important for fuel economy, but should be undertaken under expert advice.

## INSULATION OF FOUNDATIONS

It is preferable to insulate hearths and foundations, particularly around flues, but below the insulated hearth there should be provided air ventilation, the hearth being separated from the foundations by an air gap. Otherwise the soakage of heat from large furnaces into the foundations may result in the concrete cracking from expansion.

TABLE S4

Thickness of		Temperature of the hot face—° C.									
Fire-brick	Insulation	1,400	1,200	1,000	800	600	1,400	1,200	1,000	800	600
in.	in.	Temperature at interface between firebrick and insulation, ° C.					Heat loss B.Th.U. per sq. ft. per hr.				
4½	—	420	350	310	250	190	5,400	3,600	2,850	2,000	1,130
4½	4½	1,235	1,060	880	705	530	670	530	410	300	210
9	—	280	240	200	160	125	2,400	1,850	1,400	830	540
9	2½	990	850	710	570	430	790	620	460	330	230
9	4½	1,120	960	800	640	480	560	420	320	230	170
9	9	1,240	1,065	890	710	535	320	240	190	150	100
13½	—	260	220	190	150	120	2,150	1,600	1,300	740	530
13½	4½	1,020	880	735	590	440	480	360	270	210	160
18	—	180	150	130	110	90	1,030	750	600	460	340
18	4½	950	815	680	550	410	420	330	260	200	150
18	9	1,120	960	800	640	485	280	220	180	140	100

*Note:* Where no insulation is used the temperature recorded is that of the external surface of the wall.

## CONTINUOUS AND INTERMITTENT FURNACES

There is a fundamental distinction between continuous furnaces which operate for weeks or months together at a virtually uniform temperature and those intermittent furnaces which require to be cooled to a greater or lesser degree between charges, and again heated to full temperature.

Both suffer losses by conduction through the walls of heat which escapes by radiation and convection from the outer surface. In both there is required heat to raise the brickwork to working temperature. But whereas, in a continuous furnace as here defined, the heat stored in the brickwork is a negligible proportion of the total heat supplied to the furnace during its whole working cycle, in the intermittently heated furnace this heat, which is at least partly dissipated each time the furnace temperature is reduced, may be the greater part of the total heat supplied. What proportion stored heat bears to total heat depends on the time cycle of the heating and cooling operation.

In continuous or long time-cycle furnaces the insulating problem is to prevent the dissipation of heat from external surfaces. In intermittent, short time-cycle, furnaces the chief problem is to reduce the heat storage loss, whilst not neglecting to mitigate the external surface loss.

The impression widely held that intermittent furnaces which must be cooled between operations should not be insulated is erroneous. The relatively new technique of hot-face insulation provides a complete means of saving much of the very serious fuel losses that occur in intermittent furnaces through heat storage in the brickwork. This subject will be discussed in more detail.

## FUEL EXPENDITURE IN HEATING UP

When a furnace is heated up from the cold, the first stages involve the heating of the brickwork to the requisite temperature. It is frequently not realised what large amounts of heat are required for this purpose. The heat required over a period depends to a great extent upon the time cycle of the furnace. In a continuous furnace the proportion which the heat used for heating up bears to the total heat that is used in the furnace per annum is very small.

In a furnace with a weekly cycle (e.g. reheating furnaces), when the furnace is not used at the week-end it will be considerable. With a daily cycle, shutting down over-night (e.g. drop-forge furnaces and some heat-treatment furnaces) it will be very considerable. Finally, in furnaces operating on a heating-and-cooling cycle (e.g. annealing furnaces and intermittent brick kilns) in which the furnace is heated up with each charge of goods and cooled with the goods in, the heat consumption by the brickwork may assume quite surprising proportions in relation to the heat usefully consumed.

The heat stored in the brickwork in total B.Th.U. is given (Chapter V) by the product:—

$$\text{weight of brickwork (lb.)} \times \text{sp. ht. (per unit weight)} \times \text{average rise in temperature (°F.).}$$

The specific heat is within narrow limits the same for all siliceous refractories, including firebricks, silica bricks and insulating bricks. The rise in temperature depends on the conditions. The essential differences lie in the weight of the bricks. Firebricks and silica bricks weigh between 100 and 130 lb. per cubic foot in general, whereas insulating bricks are about one-quarter of these figures. Thus, for a given temperature range, the heat stored in a wall of insulating bricks is only one-quarter of that stored in a similar wall of acid refractory—an argument in favour of using as much insulating brick and as little refractory brick as possible in the walls of intermittent furnaces.

To give some illustration of the quantity of heat used for heating up the brickwork to the steady state, the conditions calculated in Table 84 have been used to calculate Table 85. Here the refractory is taken to weigh 120 lb. per cubic foot; the insulation 30 lb. per cubic foot. The specific heat of both is regarded as 0.24. The temperature is calculated from a basis figure of 60° F.

TABLE 85

Composition of furnace wall		Mean temp. in wall				Heat stored in wall B.Th.U./sq. ft. of internal surface		
Refractory in.	Insulation in.	Refractory ° F. ° C.		Insulation ° F. ° C.		Refractory	Insulation	Total
9	nil	1,250	676	—	—	25,800	—	25,800
13½	nil	1,210	654	—	—	37,400	—	37,400
9	4½	1,829	998	945	507	38,200	2,400	40,600
13½	4½	1,765	963	877	470	56,200	2,220	58,420
9	9	1,902	1,040	988	531	40,000	5,020	45,020
13½	9	1,860	1,016	945	507	58,400	4,800	63,200

## INTERMITTENT OPERATION

Clearly, the shorter the time cycle, the greater is the heat stored in the wall in comparison with the heat lost by conduction, etc., through the wall.

The first effect of external insulation is to increase the heat storage in the wall (Table 85) because the average temperature of the heavy refractory lining is increased. By putting 4½ inches of insulation on the exterior surface, the heat storage is increased by about 60 per cent. This is of no consequence in a continuous or long-cycle furnace, but becomes of great significance in a short-cycle intermittent furnace. The solution of the difficulty lies in reducing to a minimum the weight of refractory that must be heated and cooled each cycle. Until comparatively recently this solution was rendered impossible by the high

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temperature of the insulation/refractory interface, insulating materials not being available to withstand this temperature.

The newer technique of "hot-face" insulation has become possible through the development of bricks which combine high refractoriness with good insulating properties. It is now possible to construct furnaces wholly of these bricks, or line the interior with these bricks. Tables 86 and 87 indicate from published figures the value of hot-face insulation for intermittent furnaces. Table 86 (*Trans. Cer. Soc.*, XXXV, 441) shows the very much smaller quantity of heat stored in a wall constructed of high temperature insulating material. Table 87 compares the performance of an uninsulated intermittent furnace with a hot-face insulated furnace with single and double insulation.

TABLE 86. HEAT STORAGE IN RAISING HOT FACE TO 1,300° C. AND MAINTAINING TO GIVE FULL SATURATION

Type of wall	Thickness		Heat stored, B.Th.U./sq. ft.			Relative proportions of heat stored
	Brick in.	Insulation in.	Brick	Insulation	Total	
Hot-face insulation ..	9	—	—	—	15,010	1
Firebrick backed with insulation ..	18	4½	86,600	2,730	89,330	6
..	13½	4½	67,720	3,025	70,745	4-7
..	9	4½	46,950	3,310	50,260	3-4

TABLE 87. FURNACE OPERATING AT APPROX. 1,000° C. OVER 120 HOURS CONTINUOUS CYCLE WITH A WEEK-END SHUT-DOWN OF 48 HOURS  
(k = thermal conductivity in B.Th.U./sq. ft./hr./°F./in.)

Method of construction	9 in. firebrick k = 7-8	9 in. hot-face insulation k = 1-88	9 in. hot-face insulation, k = 1-88 ; backed by 2 in. insulation, k = 0-74
Loss by transmission through brickwork B.Th.U./sq. ft./week ..	146,400	39,300	25,200
Heat capacity loss, B.Th.U./sq. ft./week-end	20,090	6,540	6,810

Hot-face insulation can be used for a variety of purposes, one of which is for lining hot gas ducts. A producer gas main at a steel works which had been so lined gave the following results:—

	Before insulation.	After insulation.
Gas inlet .. ..	700° C.	730° C.
Gas outlet .. ..	460° C.	660° C.

Temperature drop along the main .. 240° C. 70° C.

The relatively low cold crushing strength of these bricks does not preclude their use for large furnaces, as is sometimes supposed. The weight of the structure is very much reduced by using the lighter bricks and the reduced average temperature in the brick for a given hot-face temperature leads to a smaller reduction in the crushing strength with rising hot-face temperature when compared with firebrick. Brick kilns up to 31 feet in diameter with a total

weight of crown of 14 tons have been in operation, using hot-face insulation, for the past eight years without major repairs ; no important part of this structure is built of firebrick.

Figures obtained on a small gas-fired forge furnace are given in Table 88. No alterations were made in this furnace other than the substitution of  $4\frac{1}{2}$  inches of insulating lining in place of the same thickness of firebrick.

TABLE EFFECT OF HOT-FACE INSULATION ON A FORGE FURNACE

	Firebrick lined	Lined with insulating brick
Time to reach 1,150° C. from cold .. .. .	55 min.	15 min.
Gas used in heating up .. .. .	400 cu. ft.	90 cu. ft.
Rate of consumption .. .. .	475 cu. ft./hr.	375 cu. ft./hr.
Gas rate for maintaining working temperature at 1,150° C. .. .. .	350 cu. ft./hr.	175 cu. ft./hr.

Finally, Table 89 (*Trans. Cer. Soc.*, XXXV, 458) indicates results claimed on a gas-fired annealing furnace operating at 750° C., in which an older furnace of normal design was replaced by a new furnace built of insulating refractory, with a non-refractory insulating backing. Part of the improvement registered was due to the provision of instruments and controls.

TABLE 89

Heating up to 750° C.	Old furnace	New furnace
Time .. .. .	4 hr.	1 hr.
Gas used per heat .. .. .	4,500 cu. ft.	750 cu. ft.
Gas used at six heats/week .. .. .	27,000 cu. ft.	4,500 cu. ft.
Saving .. .. .		22,500 cu. ft. = 83.3%
Maintaining at 750° C.		
Gas used at 44 hr./week .. .. .	33,000 cu. ft.	15,400 cu. ft.
Saving .. .. .		17,600 cu. ft. = 53.3%

The quest for high output in intermittent furnaces has led to the practices of (a) constructing these furnaces of thin, uninsulated walls to reduce the thermal storage, and (b) regarding insulation as unwise ; both practices are supposed to assist in rapid cooling and so to increase output.

These practices and beliefs are fundamentally unsound. Thin walls may be admissible for very short time cycles, but if the furnace is operated for longer than the limiting time corresponding to the thickness of the wall, these thin walls lead to a high external temperature and to great consequent waste of fuel.

Insulation—external or internal—is always an advantage even with intermittent furnaces which must be partly cooled internally between charges. The inside surface of the wall can be cooled rapidly by air (if suitable bricks are used) for the reception of cold stock, but the body of the brickwork remains hot and the furnace wall requires less fuel in the aggregate. The opinion that intermittent furnaces should not be insulated arises from ignorance of the true temperature gradients that remain in a furnace wall when the surface has been rapidly chilled.

Tables 88 and 89 indicate the much greater rate of heating up when hot-face insulation is used, as a direct result of the low heat capacity. For the same reason cooling is also accelerated, since the heat content of the brick is lower and the temperature gradient steeper.

When a furnace is closed down for a period daily or at the week-end, insulation will prevent heat losses and will enable the furnace to be started up again quickly and with little expenditure of fuel. In order that the maximum advantage may be derived from this effect, it is essential that the brickwork should be well pointed to avoid ingress of air, doors should fit tightly, and in particular the dampers must be tightly fitting so that no cold air is drawn through the furnace during the time it is closed down. Under good conditions it is possible to keep sealed-up furnaces hot for long periods. This is an alternative to banked fires or limited gas consumption that should be considered.

### RADIATION LOSSES THROUGH OPENINGS

Fig. 142 indicates a furnace in which there is a hole in the furnace wall. An

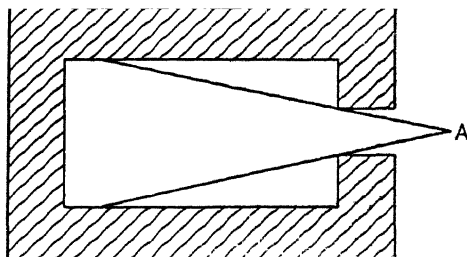


FIG. 142. Heat radiation from an opening in a furnace.

observer at A would see the interior of the furnace as if that interior were a plane surface situated at the opening and having the same area as the opening.

Since the interior of a furnace is generally taken to be equivalent to black-body radiation ( $E = 1$ ) the rate of heat radiation per unit area of plane

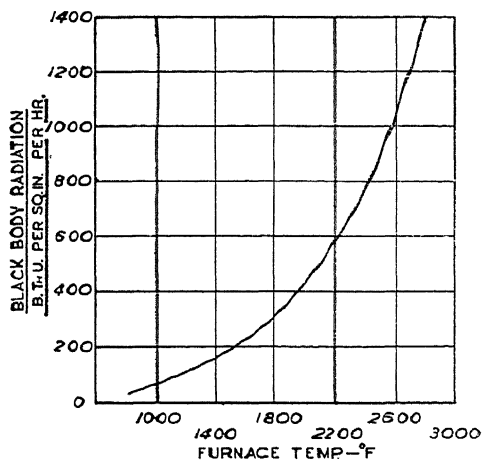


FIG. 143. Black body radiation from a furnace interior.

surface observed would be given by Fig. 143 for a furnace having walls of negligible thickness at the opening.

An example will illustrate the method of calculation. A furnace front is bricked up with a  $4\frac{1}{2}$  inch wall, through which an observation opening (peephole) of  $9 \times 2\frac{1}{2}$  inches is left open. It is required to calculate how much heat



escapes through the hole by radiation if the furnace temperature is  $2,400^{\circ}\text{F.}$  ( $1,315^{\circ}\text{C.}$ ).

Area of equivalent diaphragm =  $2\frac{1}{2} \times 9$  inches = 22.5 square inches.

From Fig. 143 black-body radiation from 1 square inch of surface at  $2,400^{\circ}\text{F.} = 800$  B.Th.U. per hour.

Heat loss from freely exposed diaphragm =  $800 \times 22.5 = 18,000$  B.Th.U. per hour.

This calculation is only correct if the furnace opening is constructed of very thin material. The result is too large because the thickness of wall obstructs the direct radiation to an extent depending on the ratio of the wall thickness to the width of opening; and it is too small to the extent to which the sides of the opening become heated by radiation received from the interior of the furnace, and re-radiate away from the furnace a part of the heat thus received. As a result of these errors the calculation just made gives figures of the order of double the real rate of loss of heat from an opening.

The figure of double just mentioned must not be taken as an invariable factor; it depends on the shape of the hole and the ratio of least width to wall thickness. Those interested in pursuing this problem are referred to the book "Industrial Furnaces" by W. Trinks (p. 111, Vol. I, 3rd Ed.). A similar calculation shows that if in a furnace operating at  $2,200^{\circ}\text{F.}$ , a furnace door measuring  $5 \times 4$  feet is left open, heat is lost at the rate of 1,728,000 B.Th.U. per hour, neglecting the factor. The factor here is 0.85, making a total loss of heat of 1,469,000 B.Th.U. per hour. The calculations are more specialised than the treatment here demands and are given not for instruction in the method of making them, but to direct attention to the importance of keeping all openings in furnaces closed as far as possible. All apertures through which the furnace can be observed should be covered with plates preferably backed by insulation.

In a furnace door which is opened periodically, continuous loss of heat does not occur, but the door lining and the sides of the opening become heated to practically furnace temperature. Thus whenever the door is opened there is the dual loss of radiation from the door and from the sides of the opening which are quickly cooled, plus the radiation loss due to the opening which has previously been calculated.

## LOSS BY FLAME OUTSIDE THE FURNACE

In order to prevent oxidation of the charge in a furnace which must be operated in a reducing atmosphere, furnaces are frequently worked under pressure. It is considered that a slight loss by gases passing outwards is less detrimental than oxidation by air leaking inwards. If, however, the furnace door does not fit well, the volume of flame passing out of the furnace may be quite considerable. The door frame will be overheated and will become warped, and the conditions will be made worse. It is difficult to estimate the amount of loss from this cause, but it is evidently a source of loss to which close attention should be paid. Maintenance of the structural work and correct fitting of doors and covers, together with adjustment to as small a positive pressure as possible within the furnace are the two ways in which this loss may be minimised. It has been concluded that with tightly fitting doors this loss may not exceed 2 per cent. of the total heat delivered to the furnace, and should not average more than 1 per cent. With loosely fitting doors, and with a flame or jets of heated gases projecting against the doors, it may easily reach 8 per cent. and at times may even exceed that amount.

This loss is especially heavy in large furnaces with ingots projecting under the door into the open, because large, half-open doors cannot be made tight. Under these circumstances it is better to brick up the opening around the ingot.

A similar loss may occur through flames within the structure not being confined to the furnace chamber. It sometimes happens when combustion is badly adjusted that the flame passes through the furnace and continues to burn in the recuperators, regenerators, or waste gas flue.

If there are adequate arrangements for recovering the heat that is lost and this heat can be put to useful purpose, the result may not be very serious. If, however, the heat is not recovered the loss may be very great indeed.

Finally, in furnaces operating under pressure, open spy-holes or cracks in the furnace wall may cause a loss of gases by flame burning in the outside air.

It has been recorded, perhaps as an extreme example, that during the inspection of furnaces in the drive for fuel economy, instances have been noticed in which there appear to be nearly as much flame burning outside the furnace as inside. With injection burners in small furnaces this condition is often caused by excessive pressure cutting down the air supply.

## SCIENTIFIC CONTROL

In any attempt at scientific control of furnace efficiency a primary knowledge of the distribution of the heat losses is essential. From the resulting heat balance it is possible to devise routine methods of control which will ensure that the controllable elements are regulated (*cf.* Chapter XXIX.).

## THE HEAT BALANCE

The main elements of the balance sheet are :—

(1) *Useful Heat Transference to the Charge.* This is readily obtained by a knowledge of the temperature, weight and character of the charge from which the heat capacity can be obtained. It entails recording the necessary information of the weights charged, and accurate pyrometry, the importance of which in their bearing on economical furnace practice cannot be over-emphasised.

(2) *Combustion and Chimney Losses.* (a) Unburnt combustible, both in the products of combustion, as deduced from gas analysis, and in the ashes derived from solid fuels.

(b) *Sensible heat and latent heat in waste gases,* obtainable from determination of the CO<sub>2</sub> content of the waste gases and their temperature. Charts constructed for each fuel as in Chapter V. facilitate the evaluation of this loss.

(3) *Furnace Structure Losses.* Heat stored in brickwork and lost to surroundings is frequently obtained by difference. Its exact determination is difficult, but considerable information of value can be obtained by the determination of external surface temperatures. Excessive temperatures, besides indicating excessive surface losses due to insufficient thickness of refractory lining, point to breakdown of insulation and defective brickwork.

## INSTRUMENTS

The instrument panel on a furnace should be the nerve centre of the installation. It should indicate at all times the rate of input of heat to the installation and whether the combustion and draughting conditions are correct. In the smaller types of plant draught gauges and pyrometers may have to suffice. The air consumption can be made an index of the heat input. Its measurement by means of an orifice gauge, a simple and inexpensive type of instrument (Chapter X.), offers a ready means of control on any type of furnace. Safeguards must be taken to avoid inleakage of cold air, which can vitiate the indications of the flow meter.

## RECORDS OF PERFORMANCE

The first essential towards attaining control of fuel economy is to keep regular records of the data necessary for watching efficiency. These include routine

recording of the fuel used, the output, the hours worked, the variations of shop conditions, and of any special tests carried out on individual furnaces. Charts of the performance of furnaces should be drawn and kept under constant observation. If the rate of fuel consumption is plotted against the rate of output generally a straight line relationship is found, which when produced to cut the fuel consumption ordinate at zero output gives the rate of consumption for the empty furnace. This represents the quantity of fuel which is required per hour, say, to bring the furnace to temperature, and maintain it in operation under the requisite conditions. Alternatively, the weekly values of the fuel ratio in terms of fuel used per ton of output can be plotted against the weekly output. Such curves give datum lines from which any improvement in method of operation or constructional alteration can be gauged and optimum rates of output assessed. A set of values for a reheating furnace is given in Table 90, from which it can be determined that a rate of fuel consumption of approximately 0.9 cwt. per hour is necessary to keep the furnace at its operation temperature, and to make good the heat losses in the furnace structure and the waste gases. The values in the last column show how the fuel ratio becomes reduced as the rate of output is increased.

TABLE 90. FUEL CONSUMPTION—OUTPUT RELATIONS IN INTERMITTENT FURNACES

Rate of fuel consumption cwt./hr.	Rate of output cwt./hr.	Fuel ratio :— Tons fuel ÷ Tons output
1.24	4.5	0.275
1.25	5.0	0.25
1.40	7.0	0.20
1.52	8.0	0.19
1.76	9.0	0.195

## PRACTICAL HINTS TO OPERATORS

The following notes comprise some brief practical hints to operators of various types of furnaces.

### *Coal-fired Furnaces*

- (1) With hand fired furnaces the depth of fire should be 15 inches. No fresh firing should exceed  $1\frac{1}{2}$ –2 shovelfuls for each square foot of grate surface. Firing too heavily causes excessive smoke and waste of fuel.
- (2) Intervals between firing should be from 15 to 20 minutes in reheating furnaces, and should never exceed 30 minutes, even in heat treatment furnaces.
- (3) The grate should be kept covered, the fires level and free from holes, and the rake used when necessary.
- (4) Firebars must be evenly spaced, and burning or warping should be rectified at once.
- (5) Unnecessary wastage of cinders in the ashes is avoided by cleaning the fires carefully. After cleaning the fire is worked back gradually.
- (6) The air supply is regulated to give bright clean fires.
- (7) Inleakage of air should be avoided, since it removes heat from the furnace and transfers it to the chimney. All sources of air inleakage should be patched up.
- (8) Dampers must be kept in good repair, and the draught in the furnace balanced to prevent either inleakage of air or blowing out of flame. In most furnaces a slight indraught below the door is consistent with balanced conditions of draught.

*Coke-fired Furnaces*

With coke firing, the larger the coke the thicker should be the fuel bed with a depth of 12–15 inches as a maximum. The conditions of firing differ from those of coal-fired furnaces, since no smoke can be produced from coke under any conditions, but guidance on all the other points is practically the same as for coal-fired furnaces.

*Gas- and Pulverised Coal-fired Furnaces*

- (1) Burners should be kept clean and in good repair. All adjustable parts should move freely. Air controls on burner injectors often become jammed through dirt, corrosion or mechanical damage.
- (2) Dampers should be free, in good working order and not subject to in-leakage of air.
- (3) The correct amount of air can be ascertained by inspection of the flame, though the indications will differ with various types of burner. More exact indication can be obtained by flue gas analysis, and it is of great help to furnace operators if a CO<sub>2</sub> recorder is installed for their guidance.
- (4) In blast burners which normally work with a non-luminous flame, a long lazy flame indicates too much gas. Correct combustion is obtained by shortening the flame until the yellowish colour just disappears. This is done by cutting down the gas, or increasing the air supply. The roar of the burner is then generally at its loudest.
- (5) In other types of burner, where mixing of gas and air is delayed, a bright yellow colour shows good combustion, the brighter the better.
- (6) Once the burners are set to give proper combustion individual burners should be adjusted as little as possible. The gas supply to the whole furnace should be reduced or increased whenever necessary by using the main valve only. As with coal-fired furnaces, in-leakage of air should be avoided whether from furnace doors, from open spy-holes or from faulty brickwork. Air in-leakage in effect transfers heat from the furnace to the chimney and is the cause of much waste.
- (7) In bogie furnaces the sand seals should be kept full.
- (8) The damper should be used to keep a balanced draught in the furnace. The correct position for the damper can be found by burning a small flame or blowing smoke in a sight-hole and adjusting the damper till the flame just flickers or the smoke just fails to be drawn into the furnaces.
- (9) Air slides and valves should be adjusted to correspond with the flow of gas. Whenever the amount of gas supplied to the furnace is altered the damper should be adjusted to re-establish the original pressure conditions within the combustion chamber or furnace.
- (10) Furnaces should not be lighted up until they are required, due allowance being made for the time necessary to attain working temperature. When furnaces are not required gas and air should be shut off and the dampers closed. The fuel supplied to a furnace should be correlated with the amount of work required by the furnace.

*Oil-fired Furnaces* (cf. also Chapter XXVIII)

- (1) The liquid fuel at the point of use must be of right viscosity if the burners are to operate properly.
- (2) Burners should be dismantled and cleaned very frequently, preferably once a shift.
- (3) Burners should be fitted with graduated valves that permit a visual indication of the amount of oil which is being used; this permits ready adjustment of oil supply to load. Similarly, dampers should be graduated.

- (4) When the furnace is shut down, the burner should be removed to protect it from the radiant heat or protected by a metal sheet. The burner should not be kept cool by leaving the steam supply on.
- (5) Water should be separated from the fuel oil; it can delay combustion and produce heavy smoke.
- (6) The sizes of apertures in burners should be checked periodically.
- (7) The general principles of efficient operation that apply to other fuels also apply to oil-fired furnaces.

*General Rules for the Economical Use of Fuel*

- (1) With existing equipment, heat treatment must be planned so that as far as possible the appliance is working at full load. Maintenance periods with no load and long periods of operation below capacity should be avoided. Major fuel economies can be achieved by careful load planning.
- 2) Flame should be kept within the furnace; gas should not be burnt beyond the furnace in the flue.
- (3) Doors should be reasonably well balanced and fit well; they should be readily operated. Flames outside the doors cause warping or damage to the door lifting gear; the damage and loss become progressively greater.
- 4) Fuel should not be burned at full rate when heats have to be held.
- (5) A record of fuel used and weight of goods heated should be kept.
- (6) Leaky brickwork is one of the greatest sources of thermal loss. The furnace brickwork should be put into first-class repair so that infiltration of air cannot occur. It should also be painted with a suitable material, e.g. fireclay and cement or for lower temperatures a mixture of tar and fireclay, to prevent air leaking in through the pores of the bricks. Attention should be paid to the condition of the furnace doors, sight-holes, etc. Furnace doors should fit closely to the surrounding brickwork. The brickwork should be examined as a matter of routine at frequent intervals and in any event not less frequently than once a month.

## CHAPTER XIX

### SPECIAL FURNACES

Structural features—Auxiliary plant—Furnace testing—Melting furnaces : hearth, crucible, shaft and cupola furnaces—Sintering furnaces—Kilns and ovens : intermittent and continuous—Furnaces for the glass industry—Reheating furnaces : soaking pits—Heat treatment furnaces : single purpose furnaces ; conveyor furnaces ; “top-hat” furnaces ; lead pots and salt baths—Furnace control—Variable heat flow in furnaces.

#### INTRODUCTION

**T**HE principles of furnace operation and design have so far only been dealt with from the standpoint of the combustion of the fuel, heat transmission and the flow of gases. The present chapter deals with the bearing of the structural features and the inferences from the heat balance on the problem of the efficient use of fuel in furnaces, and then with certain details of the operation of special types of furnaces.

These are classified on the following basis :—

- I. Melting furnaces, including hearth, crucible and shaft types.
- II. Sintering furnaces, represented as a typical example by the cement kiln.
- III. Kilns and ovens.
- IV. Furnaces used in the glass industry.
- V. Reheating furnaces.
- VI. Heat treatment furnaces.

#### STRUCTURAL FEATURES

##### STRENGTH AND DURABILITY OF FURNACES

The methods and materials used in furnace construction have an important bearing on fuel efficiency since breakdowns and delays may be a source of heat waste. Refractories and insulation have been given separate attention in Chapter XXII. The furnace framework is made either of cast-iron plates bolted together or steel structural work, iron castings and plates, selection being governed by mechanical considerations and the possibility of the part being subjected to heat and oxidation. None of these materials is satisfactory when exposed to high temperatures, and there must then be recourse to special materials such as heat resisting alloys. The development of these materials during recent years has resulted in considerable progress in the mechanisation of furnace practice, and the provision of structures and mechanisms capable of being exposed to the internal heat of the furnace. Such materials are used extensively up to temperatures of 1,000° C. and for special work at even higher temperatures.

They are required to be resistant to scaling, and to have sufficient hot strength to be capable of being used in many types of mechanisms such as conveyors. The factor on which selection of the material is based in regard to this last-named property is known as the creep strength, that is the load which the metal can carry for extended period at the temperature involved without perceptible creep. Other properties of significance are resistance to carburisation (carburising boxes), to the effects of repeated heating and cooling (chain conveyors, particularly those immersed in quenching media), heat diffusivity (glass furnaces), malleability (for bars and sheets), machinability and, in short, what is important is whether the materials can be cast or fabricated into the many forms to be found in the modern furnaces.

An extensive literature of the subject exists to which reference may be made.

The design of the structural framework of the furnace is based on the well-known principles of mechanical construction. A special point in its bearing on fuel efficiency is that robustness is essential since repeated heating and cooling of structural members together with the concomitant expansion and contraction of the refractory materials used can induce, in course of time, a remarkable degree of distortion of framework. Large furnaces of insufficient robustness have been known to be several feet out of alignment after extended use where sufficient strength and allowance for expansion have not been applied. Then cracks in brickwork more readily arise, resulting in air leakage or flame emission, and a lowering of furnace efficiency.

## THE FURNACE CHAMBER AND ROOFS

Provided the abutment of the arch, or skewback, is adequately supported by a strong framework with tie rods, the arch on heating must rise. In selecting brickwork the expansion characteristics of the refractory to be used must be known, also whether there is any after-contraction on firing to the temperature at which the roof is to be operated. If the brickwork were incompressible then with good shaped material, the bricks would be in contact over their full length near the arch support, but near the crown of the arch they would tend to touch only at the inner edge. Proposals have been made and roofs are constructed of specially shaped end bricks abutting on the skewback to allow for this variation to be taken up during heating. Actually, all firebricks are capable of crushing elastically to a degree compatible with a robust construction, and, in general, a rigidly bricked roof, provided ample rise in the arch is allowed (for example,  $1\frac{1}{2}$ -inches to the foot), gives satisfactory service. Further, the softening of the bricks at furnace temperatures and the lower refractoriness resulting from the application of pressure, permit plastic compression of the bricks on the furnace side of the arch, and tend to equalise the load and compensate for irregularities of shape. Again, the line of thrust of a hot arch departs farther from the median line of the brickwork towards the hot surface as the centre of the span is approached, so that the greatest deformation is likely to arise where the initial separation of the brickwork is greatest.

Furnace roofs fail either by yielding of the skewbacks, or by spalling and wear. Spalling can weaken an arch, and it can then finally be reduced to a condition of collapse through normal slag attack and erosion by furnace gases and fume. It is remarkable how thin a roof can wear without collapse provided the abutments are rigid, but then the question of heat loss by conduction and from external surfaces possibly outweighs the loss of durability of the roof, unless as so frequently occurs in industrial practice, it is necessary to wait for a convenient period for repair. It may be urged with good reason that planned maintenance requiring repairs to be done at specified intervals is preferable to the policy of striving for the absolute limit of durability.

Under suitable conditions abutments can be made rigid and dependence placed on the rise of the arch; alternatively, spring loaded or adjustable tie rods may be used, but these latter require skill in adjustment, and unskilled handling may result in failure of the arch.

Distances greater than 18 feet are seldom spanned by a single arch. They are either broken up into several smaller spans, supported on skewbacks, carried on longitudinal girder reinforcement, or slung from overhead beams. Flat roofs, built of refractory tiles suspended by hangers from overhead girders are also used. The relative merits of the sprung arch and the suspended roof depend entirely upon the conditions, the occurrence of movement, as in soaking pit lids, being a factor of significance operating against the success of the flat

roof. A complicated shape constructed in fired refractory material is always more vulnerable to the effects of variable stresses than the simple shapes capable of being used in an arched roof. On the other hand, under suitable conditions of use the suspended roof may be preferred since it eliminates the need for the skewback and gives freedom of design in the sidewalls for the provision of openings to the furnace chamber.

Roofs 9 inches thick are generally insulated with advantage, except in special cases over combustion chambers where the temperature conditions are too severe for the brickwork. Selection of a better-class refractory brick may, on the other hand, be here successful.

Roofs may collapse from mechanical injuries arising from explosions or impacts. The best of brickwork and design is of no avail against destructive treatment which may arise from errors in lighting up or in controlling charging operations.

In roof construction attention should be devoted to the choice of brick shapes to give uniform stresses, and a reasonable gas tightness to obviate the excessive leakage of hot gases which occurs in a badly built roof due to the buoyancy of the hot gas. In certain types of kilns it is considered good practice that the crown or arch should spring from a wall behind the kiln lining. The construction of crowns of certain types of kiln differs in important features of technique from that of other high temperature furnaces, and being a special field of practice is here excluded.

## SIDE WALLS AND HEARTHES

If the side wall forms the support for the roof its correct construction is of primary importance both in relation to the thrust of the roof and of the hearth. The major fault to be avoided in the construction of the lower part of the furnace lining is a lack of adequate strength for the supporting girders of the hearth, which should be securely anchored to the buckstays, whilst side-wall support in high temperature furnaces should be provided by the use of strong cast-iron plates, particularly in situations not required to be made accessible for the purpose of brickwork repair. Such construction admits of the ready use of the highest grade insulating bricks in side walls.

Adequate insulation of hearths in modern furnace practice is imperative, and it is practicable to obtain material capable of withstanding the heaviest hearth loads. Recent research has shown that economical furnace operation depends to a considerable degree on the manner in which uniformity of temperature is rapidly attained. Since, except in under-fired furnaces, the hearth is the coldest part of the furnace, provision for its adequate heating is a prime factor in good furnace design. At the same time adequate refractoriness, stability and resistance to attrition must be provided.

An excessive amount of brickwork does not constitute good practice. There must be the correct combination of suitable refractories to maintain thermal insulation and mechanical stability. Stability is assisted by ventilation, i.e. the most satisfactory hearth is mechanically supported and air-cooled from below.

Slagging troubles or soft hearths can usually be overcome by a correct selection of refractories, for example by the use of a superior firebrick, or a basic refractory such as dolomite, chromite, or in special cases magnesite, and in the rarest sillimanite, carbon or carborundum. In siliceous hearths slag is run off in liquid form through a tap hole, and the condition of the hearth made good by fettling with a suitable refractory as, for example, sand or iron oxide cinder when the slags are ferruginous. Dry hearths should always be kept clean and free from scale and cinder by raking at convenient intervals of time.



## FOUNDATIONS

The design of the foundations is not a feature that can be usefully discussed as a factor entering into the problem of correcting practice in normal operation, since it is a feature of the initial construction and can only be adequately met at the installation of the furnace. Overheated foundations may, however, be the source of uneconomical operation by permitting subsidence which gives rise to cracks in brickwork, and resultant wasteful inleakage of cold air. Thin, non-ventilated, brickwork hearths which are, however, capable of being readily altered, can cause overheating of concrete, dehydration of which commences at 260° C. and is completed at 480° C. Limiting thicknesses are  $D/6$  for furnaces working at below 1,205° C. and  $D/8$  for 870° C.,  $D$  being the shortest dimension of the hearth in feet.

## HEARTH SUPPORTS, SKIDS AND HANGERS

A wide range of mechanical devices is used for the purpose of hearth supports, skids and hangers, to enable stock to be moved readily through continuous furnaces. From the standpoint of fuel efficiency certain general principles should be observed in their construction and maintenance.

Apart from waste of fuel and output arising from interrupted campaigns and the preservation of general economy of operation, excessive heat losses may arise from water cooling, but its use under certain circumstances is inevitable. Skid pipes are usually of hydraulic section, and must be securely anchored in the furnace hearths. They may be fastened into the anchorage in such a manner as to admit of being turned to expose new and unworn surfaces for further service or, alternatively, welded strips may be used to take the wear. Safeguarding the water supply and preventing blockages in pipes are almost self-evident precautions.

The main objection to the use of water cooling in skids arises from cold patches in the stock, and finishing periods of heating must then be made by transfer to solid refractory hearths. The use of dry skids, hearth plates, and mechanical conveyors introduces a field of furnace practice of a specialised type. Where mechanical conveyors are used the amount of sensible heat lost from them may be an important item in the heat balance. The heat-resisting steels already mentioned are frequently used for these purposes.

## JAMBS, DOORS AND OPENINGS IN SIDE WALLS

Openings in side walls can be a source of inefficiency if robust methods of construction are not applied. This applies particularly to furnaces operating at temperatures above 1,000° C. Open doors lose heat at a very high rate from direct radiation to the atmosphere. Distorted buckstays, piers and jamb reinforcement give rise to air inleakage, and flame loss, for proper fitting of doors is then no longer possible. Modern refractory practice in the provision of high temperature insulating materials, on the other hand, makes practicable more robust and better doors. A most important feature of furnace practice is attention to the sealing of the doors to prevent such leakages. The main factors in overcoming such disabilities are the initial robustness of the reinforcement and the use of well-fitting door frames, side plates and steady mechanical action during movement. Water-cooled doors may be essential, but the heat losses due to the cooling action should always be examined in relation to the general economy.

A badly fitting discharge door on a reheating furnace having a gap along its top edge 1 inch wide and 24 inches long will give a heat loss of approximately 200,000 B.Th.U. per hour due to escaping gases if the furnace is operating at

a temperature of  $1,100^{\circ}\text{C}$ . with an exhaust temperature of  $700^{\circ}\text{C}$ ., and with a normal furnace pressure. This clearly shows the loss which can be obtained from a comparatively small opening.

### AUXILIARY PLANT

In addition to the furnace proper there is often much equipment ancillary to the furnace which requires attention if the best results are to be obtained.

### DAMPERS

The function of a damper is to regulate the draught, and accordingly it becomes the **prime** instrument of the furnace economy. The controls should always be situated in such a position as to be readily accessible, and preferably also permit of sighting the flame when adjustments are being made. The choice of the material from which the damper is made depends upon the conditions of operation. It must not crack or warp, nor must brickwork remain dislodged. It should move easily in its seating and be readily capable of fine adjustment since quite a small movement may mean a waste of many tons of fuel. Allowance should be made for expansion, and suitable covers or leakage seals provided to prevent the inflow of cold air to the flue. If the furnace is temporarily shut down the damper should always be closed. Closing the damper when shutting down the furnace is equivalent to turning off the fuel valves since it prevents cold air leakage.

Water-cooled dampers of special design or dampers constructed of heat-resisting alloys are necessary for the highest temperatures. For gas-tightness they should be made to slide on inclined damper frames machined on the working face.

### FANS

Controlled draught, whether "forced," "induced," or "balanced" (Chapter VI) assists the maintenance of the correct air-fuel ratio, and the adjustment of the rate of heat input to the most economical value. Mechanical draught is generally indispensable with recuperators, which require high velocities of air flow for adequate efficiency, with consequent increase of resistance.

Fan design and selection is a matter for the specialist. The principal consideration when fans are installed is to see that they are maintained efficiently by regular lubrication and cleaning. Particular care should be paid to the inlet side of the fan and suitable protection placed to prevent foreign matter being drawn into working parts. With induced draught fans disastrous results may arise from flame impingement on the blades or overheating from other causes. Valves and dampers in the fan circuit should be readily controllable. A fan gives a ready means of registering the air flow to the furnace either by pipe attachment with orifice gauge placed at the air inlet or by the use of an orifice gauge in the supply main (Chapter X). Care should be taken to ensure that the character of the flow in the main is suitable for accurate measurement if this method is used. Straightening grids may be used in difficult conditions to prevent swirl effects.

### CONTROL VALVES

Control valves, as their name implies, regulate the amount of fuel entering the furnace. For every furnace there is an ideal rate of consumption of fuel which is required for the particular heating operation. Any departure from the correct setting of the valve results in loss of efficiency. In intermittent furnaces there is reached a moment at which the soaking stage is being approached, and the rate of heat input must be reduced. It is accordingly imperative that control valves should be provided with suitable indicators, for

example quadrants and pointers, to show the exact setting of the valve. All valves should be examined, cleaned and adjusted regularly to ensure that they are in good working order.

## BURNERS

The accurate distribution of temperature is a matter of importance, frequently calling for the application of gaseous and liquid fuels of all types. The burner construction is accordingly a critical feature of the furnace design.

The main distinction of the various types of burners in common use lies in the method of aeration used. The intensity of heating and the form of the flame are governed by the nature of the fuel used, but for any given fuel the nature of the flame and the character of the heat release is controlled by the manner in which the combustible and the air are brought together. The flow characteristics of the two streams are dependent upon the laws of flow of fluids, but in the flame a complex system of changes may be brought about, and knowledge of the circumstances affecting the heat release is so far more or less empirical.

*Natural Draught Burners.* This type of burner is essentially an injector burner of the bunsen type, the gas being slightly above, and the air at, atmospheric pressure. A proportion of the air is entrained with the gas as primary air, the remainder being induced by the furnace draught as secondary air.

*Preheated Air Burners.* This type takes the form of one or more ports, and is supplied with preheated air from a heat recovery appliance, a recuperator or regenerator. It is usually used on large furnaces. Burner controls may consist of slide dampers built into ducts both in the air and gas supply systems. The gas is delivered at low pressure, and the air either by natural draught or at low pressure after passage through the preheater. The method of draughting the furnace and of controlling the waste gas dampers has a material influence on the nature of the flame.

*Low Pressure Blast Burners.* With these burners the gaseous fuel is supplied at pressures between 2 inches and 8 inches w.g., and with air pressures varying from a few inches to 5 or 6 lb. per square inch. They may be used with either cold or preheated air or both. They constitute the most simple and general type. The gas and air supplies are generally separately controlled, and it is advisable to ensure that the control valves are fitted with suitable quadrants and indicators to assist in the correct setting of the valves. They are normally fitted into a cone-shaped refractory burner block, which serves the multiple purpose of assisting the flow of gases to give the correct form to the flame, of assisting ignition at lighting up, and of maintaining combustion by the proximity of a hot radiating surface. With a refractory burner throat the relatively cold flame of a low grade fuel is not so readily blown off the burner. The maintenance of the correct gas-air ratio may be assisted by coupling the air and gas valves together through a proportioning mechanism, but it is essential to have good pressure regulation, as by a pressure governor on the fuel gas. Turbulence and good mixing are assisted by providing deflecting vanes in the annular throat carrying the air to give a vortex motion to the gases.

Crude gas is burnt in both of the above types of burners, but ample capacity must be allowed in the gas passages and valves, where tarry deposits may accumulate and cause stoppages. With hot gas, good insulation of gas supply mains is essential to keep tarry constituents as far as possible volatile, and provision should be made for the draining away of the tar without interruption of operation. Adequate cleaning gates, requiring the minimum of time for clearing deposits, are necessary. Bends should give good streamlining and mixing devices and nozzles should not be used inside the burner. Gas and air should have an ample mixing space. Disc-type valves are preferable for use with gases likely to contain tarry or dusty constituents.

This type of burner readily permits adjustability of flame length by admitting the air in separate supplies, an inner and an outer stream supplied in an axial direction.

The so-called diffusion burners fall into this class. In these the gas and air are made to diffuse slowly into one another, whereby mixing is gradual and the maximum length of flame attained. The ports may be annular, one within the other, when the flame is circular, or they may be long slots producing a flat stream of flame. They are particularly suitable for plate heating when operating on mixed gas.

It is frequently overlooked that air preheat changes the character of the flame, and that where the temperature of preheat is varying a specific setting of a flame cannot be maintained because the air is constantly changing in volume. Accordingly, burners utilising preheated air should be good mixing burners.

*High Pressure Gas Burners.* This type of burner may be operated in between one and three stages of injection according to the class of fuel and conditions of use. Gas pressures may be from 1 to 10 lb. per square inch. Alternatively, a similar type of burner may be operated by means of pressure air used as an injector to entrain gas supplied at low pressure.

With the high pressure burner the fans are not required, distribution mains are reduced in size, and steady pressures can generally be maintained by the use of a reducing valve as governor, though successful installations are operated without a governor. This system is convenient for use in converting existing furnaces to gas firing in locations in which space is limited, and only the simplest attachment to the furnace structure is permissible. With the elimination of air pipes and fans the risk of explosion is eliminated. The burner, however, lacks flexibility. The air/gas ratio is influenced by the pressure in the furnace and the output of the burner, and the more these factors vary the greater the departure from ideal combustion conditions. To avoid backfiring the lower limit of output is about  $\frac{1}{3}$  to  $\frac{1}{2}$  of full load, so that the burner is not suitable for furnaces requiring to be operated in a wide range of temperature—as, for example, some types of tempering furnace. The pressure of gas required to be available in the burner is less with the lower grade gases on account of the higher gas/air ratio, the required pressure usually being :—

	Calorific value (gross)	Pressure in. w.g.
For coke oven and town gas .. ..	500/550	118/196
Mixed gas (coke oven and blast furnace) .. ..	200/240	59/118
Producer gas .. ..	150	39/59
Blast furnace gas .. ..	95	20/39

Single-stage induction can be applied up to the point at which the air requirements of the gas do not exceed 2.3 times the gas quantity. The gases of lower calorific value have a lower gas pressure at which the flame flashes back in the burner, and the lower limit of capacity lies between 25 and 50 per cent. of the highest load according to the type of fuel gas and the furnace temperature. Since coke oven gas and town gas require more than 2.3 times their volume of air two-stage induction is generally applied when burning these gases.

Appliances for the burning of town gas and coke oven gas as a furnace fuel are described in Chapter XXVII.

*Pre-mixing Burners.* This is a popular type of burner which admits of the gas and air being delivered to the throat ready mixed. It has the advantage that the combustible mixture can be distributed by suitable mains to a number of different points in the furnace to produce the required remy distribution. The aeration of the gas is effected in an injector mixer by the use of high pressure air, the gas pressure being stabilised by the use of a governor.

This burner is suitable for a very uniform furnace atmosphere, steady heating conditions and a well distributed supply of heat. Alternatively, high pressure gas can be used to induce air.

The velocity of the gas in the distributing mains must be above the velocity of flame propagation, and at the burner inlets to the furnace cooling fins are used to safeguard against backfiring. The arrangement is unsuitable for high degrees of preheat of gas or air. The safe limiting temperature of air preheat is  $300^{\circ}\text{C}$ ., though it has been claimed to be practicable to  $500^{\circ}\text{C}$ .

With gases of low calorific value there is a greater tendency for the flame under normal operating conditions to be blown off the nozzle of the burner. These burners have the advantage of simplifying control from the standpoint of the operative. They are readily adaptable to automatic regulation of furnace temperature.

The principle of pre-mixing may also be combined with the use of secondary air at the burner nozzle, either supplied by means of a fan or induced by the draught of the furnace. The use of secondary or tertiary air can be applied in this manner to any type of injector burner. It is readily applied with high calorific value gas because the mixture ratios with such a fuel are appropriate and far above the upper explosive limit, but as each additional stage of air admission is made there follows the attendant loss of simplicity of control.

*Combustion in Films and Surface Combustion Burners.* Finally, there are the multi-jet burners in which the gas and air are subdivided into a number of small streams, producing in effect a series of films of flame. The flame volume is controlled, and burning of stock obviated. Excessive local temperature in the body of the flame is also avoided.

The modern burner is virtually a multi-holed brick, the refractory surface of which becomes highly heated and so forms an effective radiator. In the latest type the surface of the refractory is formed in the shape of a hemispherical cavity which serves as a concave reflector of radiation and accordingly is a very efficient heating appliance. The ultimate result of the combustion of gases in contact with refractory surfaces is the production of flameless incandescent surface combustion. Air preheating is practicable. The burner brickwork must be made of a high grade refractory; the fashioning of the burner must be of a high standard of workmanship and accordingly the normal question of care in use and adequate maintenance introduces an operative factor which may be a disadvantage in certain applications.

*Control of Furnace Atmosphere.* In furnace operations the maintenance of a specific type of furnace atmosphere is frequently essential in order to avoid oxidation of the stock or to preserve a surface condition (see Chapter XVIII). In the more specialised operation such as bright annealing the material may be heated out of contact with the products of combustion, and controlled atmospheres provided by means of an auxiliary appliance. The type of atmosphere used depends upon the class of material heated and the operation involved. The matter is one for specialised study since in the reactions involved many complicated equilibria must be considered.

In fuel fired furnaces for the normal method of maintaining a constant furnace atmosphere with varying rates of fuel supply a constant fuel/air ratio is required. The commonest method of adjusting the furnace atmosphere is that of visual inspection. With some fuels the decision is relatively easy; a so-called reducing atmosphere is smoky and a haze may be apparent; a neutral or oxidising atmosphere is clear. With clean gaseous fuels, coke oven gas, water gas or clean producer gas these indications are absent. With certain types of burner the brightness of the flame is a guide, but generally in the absence of any positive means of controlling the fuel and air flow resort must be made to gas analysis.

With gaseous fuels the use of automatic proportioning equipment is the simplest and the most reliable method of control. Mechanical interconnection of gas and air supply is practised, but is not infallible, since with variations of gas and air pressure, fuel quality and furnace draughting the settings of the valves may readily be thrown entirely wrong. Other devices used are :—

(i) Proportional mixers, operated by a positive blower, a fan or a jet, working in conjunction with a gas governor to maintain constant pressure of gas supply. A rich mixture is induced through adjustable ports and secondary air is induced at the burner nozzle. In another type an inspirating Venturi tube acts both as pump and proportioning device. Either gas or air under pressure may serve as the power medium.

(ii) Gas/air controls, applied by means of flow orifices. The flow of gas and air is measured by means of orifice gauges. The pressure head across the gauge is proportional to the square of the velocity of flow for fluids of low viscosity. The pressure drop through the two orifices by acting on a balanced system of diaphragms or floats sets up a pressure differential, which operates a relay either one way or the other to operate the power unit. The power unit operates the control valve governing the variable flow. By setting the differential mechanism by a convenient device the gas/air ratio can be adjusted to any desired value.

*Rules of Practice for Burner Operation.* It is an important feature of all burner efficiency that certain simple rules of practice should be observed in their operation. For brevity they are summarised as follows :—

(i) Correct setting of the gas and air valves, and the adjustment of the damper to give balanced draught in the furnace is the first golden rule. These settings should be marked on appropriate quadrants.

(ii) Where several burners are operated from one rail the valves on each should be adjusted to give correct heat distribution and the main gas and air rates are controlled on main valves and dampers.

(iii) The installation must be operated at the correct pressures. Pressure gauges should be used.

(iv) Good maintenance comprises adequate cleaning, changing of corroded or damaged jets, free valves, concentric jets or burner throats, correct seating of the burner throat in the refractory port.

(v) Governors, proportioning apparatus and valves should not be so placed as to become overheated.

(vi) All instruments are to be kept clean and in good order. Makers' rules should be observed.

## LABOUR-SAVING APPLIANCES

In modern furnaces many new types of mechanical appliances are now becoming an essential feature of the furnace operation, including charging machines, mechanical stokers and the driving mechanism for rotating hearths. Charging machines should be flexible, easily controlled, simple to adjust and quick in their action. In the use of conveyors, temperature must be controlled within specified limits to avoid destruction by overheating. With moving hearths sand seals must be always kept properly filled with suitable material; otherwise damage to bogie carriage side plates and wheels or serious inleakages of cold air may occur. Foreign matter must not be allowed to drop into the seal to cause excessive friction or dislodgment of brickwork.

## IMPROVEMENT OF EFFICIENCY AT FURNACE REBUILDS

Certain salient features have now been discussed relating to the bearing of the furnace construction on general efficiency. Matters of control, such as

adjusting combustion to give the correct mixture of fuel and air, and dampers to permit of the proper setting of the draught, or loading the furnace to the economical limit are all matters of day to day practice, which do not in any way affect the need for modification in the furnace structure. In course of investigations into conditions of operation it may become apparent that improvement can be effected by rebuilding or structural change in the furnace. Counsel to pay attention to possible changes in these respects must be tempered with the advice that since the science of furnace heating involves many complex factors, it is desirable to take the advice of a furnace expert on structural changes.

Attention should be paid to the maintenance of mechanism and furnace brickwork. The cleaning of flues, repair of dampers, maintaining the cleanliness of heat exchange surfaces as in recuperators, and the sealing of cracks in brickwork, all have an important influence on the amount of fuel consumed. An unsuitable size of grate, a furnace too large because it has been built for over-size material rarely handled, crowns too high or too low, flues of insufficient capacity or defective design, unnecessary burners, wrongly disposed and dimensioned ports, fans of an uneconomical capacity, unsuitably dimensioned combustion chambers, insufficient lagging of doors and furnace structure, are all capable of being corrected during the course of normal maintenance. Many of these defects are to be found even in modern furnaces because users are still disposed to look only at first cost. A robust, well-built and insulated furnace is always the cheapest in the long run.

### INSTRUCTION BOOKS

Finally, in furnace or kiln operation, there should be compiled at the installation of the plant a handbook of guidance to the plant operators and engineers, in which should be specifically stated the essential "rules of practice" for operating the plant, and the points to be watched in maintaining the plant in good order. Such handbooks are now prepared by the best constructors and, if consistently used by successive operators, are most valuable instruments in the maintenance of efficient practice. They should be adequately illustrated by simple sketches, and expressed in brief and direct terms.

### FURNACE TESTING

A number of British Standard Specifications have now been prepared for the determination of the performance of various types of furnace. These give guidance as to the technique to be employed for the measurement of fuel consumption, throughput and heat losses. The methods to be employed for the determination of the heat balance are indicated. The interpretation of the results, however, is quite another matter.

### INFERENCES FROM THE HEAT BALANCE

The normal method of control of heating furnaces is based on the use of graphs of fuel consumption and rate of output. (See Chapter XXIX.) In order, however, to establish improved practice by modification of the furnace operation or design, detailed analysis of the heat balance is necessary.

Since—

heat supplied = useful heat + structure loss + exhaust gas loss,  
it is apparent that for a constant fuel consumption and a constant structure loss an increase of useful heat can only take place by reduction of the exhaust gas loss. The value of a knowledge of the temperature and composition of the gases leaving the furnace accordingly becomes apparent.

In batch or intermittent furnaces, since the temperature of the furnace

passes through a cyclical change both the structure loss and the exhaust gas loss vary and there must be added also a knowledge of the storage and external heat loss.

## STRUCTURE LOSS

The structure losses become more important the higher the furnace temperature, and heat protection is then all the more necessary. Whether inner linings can be replaced by hot-face insulation or first quality brickwork insulated by standard external insulation is mainly a question of temperature, and amount of wear and attrition imposed by the process on refractories. Furnace crowns over fireboxes operating reheating furnaces cannot be insulated unless high-grade refractories are used. Insulation may only result in uneconomical refractory wear since the mean temperature of the inner lining is raised by the use of the external insulation. In pulverised fuel firing the reaction of possibly molten ash with refractory material imposes a limitation on the method of building. Expedients to overcome any disabilities of this kind include the provision of patched linings of dolomite, magnesite or siliceous materials in combustion chambers, and the construction of flue off-takes of ample size and accessibility to remove accretions of ash and clinker.

In interpreting the characteristics of the refractory it should be borne in mind that the safe limit of under-load refractoriness is in general 100° C. lower than the value shown by the normal laboratory test, since at high temperatures refractories tend to behave as semi-viscous bodies, and with the passage of time tend to flow, so that the structure may collapse. Alternatively a greater thickness of refractory brick may be used which will reduce conduction loss and at the same time give greater stability to the structure. This technique applies in the use of both suspended and arched crowns.

Artificial cooling either by means of water as in jambs and partition walls or compressed air leads to substantial losses. Except in exceptional circumstances compressed air is a wasteful and expensive means of controlling the flow of heat. The temperature of the external surface of a 4½ inches firebrick wall can be reduced from 400° C. to 40° C. by means of a jet of compressed air, and the temperature gradient through the wall when it is maintained at a temperature of 1,400° C. accordingly increased from 1,000° C. to 1,360° C., thereby causing an increase of 36 per cent. in the heat transfer.

The storage loss has been discussed in Chapters XVIII and XXII. This loss and the wall loss by conduction are complementary for the lower the storage loss brought about by reducing the thickness of the wall the greater the conduction loss. The relative merits of each case must depend upon the period of the heating cycle, whether the heat stored has time to penetrate the wall before the heating operation is completed. To determine when the loss by conduction overtakes the loss by storage it is noted that equilibrium of heat

conduction begins to occur when  $\frac{L^2}{4Dt}$  approaches 0.4,

where  $L$  = thickness of the wall (metres)

$t$  = time since the beginning of the alteration in temperature, hours

$D$  = thermal diffusivity, sq. metres/hour.

## EXHAUST GAS LOSS

In all furnaces the use of the continuous principle is conducive to a low exhaust gas temperature, but in batch furnaces the gas must leave the furnace at a temperature above that of the stock. Particular care must accordingly be taken to obviate air leakage. How far this temperature must be above the temperature of the furnace depends upon the character of the heat transfer



from the gases. Reference to Chapter VIII will show that in this type of furnace the governing factor is gas radiation, which outweighs the influence of convection. Gas radiation, apart from the composition of the gas, depends only on the thickness of the gas layer and the temperature. Since the temperature is fixed by the operating conditions it must be to the advantage of the furnace economy that there shall be within it large radiating masses of gas. This involves the use of relatively large furnace chamber capacity, providing that such provision is not at the same time outweighed by an increased structure and external surface loss, and that layering of the hot gas below the roof does not give a cold layer of air adjacent to the charge. Such a cold layer of gas forms an insulating layer since it is impenetrable to the dark heat rays of the hotter gas above. Accordingly the expedient of raising the crown of a furnace to give a better radiating gas layer and an increased surface of direct wall radiation will fail if care is not at the same time taken to ensure complete gas circulation and prevent air infiltration. A ribbed roof construction to create turbulence, as well as to increase heating surface, is a beneficial form of construction in this respect. The use of auxiliary burners and turbulent admission of secondary air are also expedients used to effect the same advantages.

Heat transmission coefficients in industrial furnaces of the continuous push type vary from 50 B.Th.U. per square foot per hour per °F. at 2,200° F. to 20 at 1,480° F.\* for gas layers of about 1 foot 6 inches in thickness. The reduction of the heat transfer coefficients with almost the fourth power of the gas temperature produces an economic limit to the lengthening of the chamber of continuous furnaces. It may then be more economical to use the metallic recuperator to preheat the air for combustion, in which the advantages of convection transfer under conditions of forced circulation give a better utilisation of heating surface.

Summarising the implications of this discussion of the heat balance it may be concluded that fuel efficiency is promoted by the following :—

- (i) The provision of adequate heating surface.
- (ii) The largest practicable space for combustion and the heating gas layer.
- (iii) The prevention of cold gas layers by means of auxiliary apparatus for creating turbulence.
- (iv) Prevention of infiltration of air by making the furnace as gas tight as possible and by control of draught.
- (v) A critical rate of throughput of material to be heated which is determined by the point at which the combined losses due to the structure absorption and the sensible heat in the exhaust gases are at a minimum value. This is consistent with a good performance in regard to the data discussed later.

## EXHAUST GAS LOSS BY UNBURNT COMBUSTIBLE

It has already been indicated that unburnt combustible can result in serious loss of furnace efficiency. Thus in the combustion of blast furnace gas 5 per cent. CO + 1 per cent. H<sub>2</sub> in the waste gases is equivalent to a loss of 25 per cent. of the potential heat of the gas, and 1 per cent. CO is equivalent to an increase of about 90° C. in the flue gas temperature.

Combustion control can be assisted by the use of secondary and tertiary air introduced at positions in the flow of the gas stream, provided that the change is not endangered by excess air or a flame too strongly reducing (i.e. below a temperature of 900° C.). The same effect can be attained by auxiliary burners carrying the needed excess of air. Other effective measures are the use of air-gas controls.

\* Metric equivalents approximately 250 Kg. cal./m<sup>2</sup>/h/° C. at 1,200° C. to 100 at 800° C.

For certain types of operation where low temperatures and good temperature uniformity are required long flames are used involving slow diffusion of volatile combustible and air. These are the most difficult to control and without late additional air admission may be subject to the disability of local overheating.

### LOSSES UNACCOUNTED FOR

In the heat balance sheet there is always an item to cover the unascertained losses which are not readily measured, and the balance of error in making the determination of each specific loss. These include :—

- (i) Loss in foundations.
- (ii) Gas leakage.
- (iii) Losses from open doors.

(i) *Losses in Foundations.* The loss in foundations is a somewhat special study, which has been taken up by J. D. Keller (*Amer. Soc. Mech. Engrs.*, 1927-8, 49-50. F.S.P. 50-57). The total flow of heat through hearths of a given shape was found to be proportional not to the area, but only to the diameter or width. The conception of an equivalent thickness of hearth was taken to be that thickness of a wall of the same material, through which the rate of flow of heat would be just the same as the actual rate through the hearth if one surface were held at the temperature of the interior of the furnace, and the other at the temperature of the surrounding air.

In unventilated hearths the equivalent thickness of the hearth at its centre was found to be slightly more than half the least width of the hearth. The average equivalent thickness referred to the whole area of the hearth was one-quarter the hearth diameter for circular furnaces,  $22\frac{1}{2}$  per cent. of the width for square hearths, and its value increased as the ratio of the lengths of the sides of rectangular furnaces, approaching a limiting value of 27 per cent. of the least width for very long hearths.

Formulae have been proposed for determining the temperature penetration into the ground. From them can be deduced the continuous nature of the heat saturation, which can lead to failure of foundations, the risk of which is not entirely eradicated by insulation. That is to say, insulation only delays the heat penetration ; it does not stop it. Accordingly high temperature furnace hearths should always be ventilated.

(ii) *Gas Leakage, and (iii) Losses from Open Doors.* In reheating furnaces the gas pressure difference produced in a column of hot gas due to its buoyancy as against the same height of the outer atmosphere amounts to approximately 0.012 inch w.g. per foot (1 millimetre w.g. per metre) height. This buoyancy effect of hot gas must accordingly produce a positive pressure at the top of a furnace door, if the gas pressure at the sill level is to be zero, and no air be allowed to be drawn in. Alternatively, to balance the draught to give zero pressure at the top of the door is equivalent to carrying a negative pressure, that is an indraught, at the sill level. Thus, when the door is opened either an indraught of cold air or an outblast of hot gas results. Generally both occur.

Further reference to the magnitude of the radiation loss from hot surfaces (Chapter VIII) permits an estimate to be made of the direct heat loss derived from this source. Accordingly the heat losses due to the repeated opening of furnace doors may be considerable, and attention to this phase of the operation is imperative (Chapter XVIII). The importance of gas tightness and correct balancing of the draught is again apparent.

Having thus discussed the general principles of furnace construction and operation, attention will now be given to specific types of furnace.

### I. MELTING FURNACES

Melting furnaces may be of hearth, crucible or shaft type. The technique of operation invariably depends upon special technical considerations, peculiar to each process, and frequently these have an over-riding influence on the manner in which the melting operation is conducted. It is therefore only possible here to review those features which are common to melting furnaces as a class.

Hearth and crucible furnaces for those processes requiring the highest temperatures, for instance steel making and glass manufacture, are frequently combined with regenerators or recuperators. In shaft furnaces the fuel is mixed with the charge, and the air for combustion blown in at tuyeres.

Apart from problems of mechanical design, economical performance depends upon three major features :—

(i) The design of the melting chamber, its shape and dimensions in relation to the rate of melting desired, and a critical rate of heat input up to the limit at which the refractories are able to maintain the mechanical stability of the furnace. Accordingly, the search for economy must be made by the study of the dimensions in relation to the rate of output required, and determinations of the critical rates of heating necessary for each set of conditions ruling. This means that importance must be attached to surveys of draught conditions and methods of firing.

(ii) Efficient combustion is of paramount importance since high rates of melting require the maintenance of certain flame temperatures, and whether conditions are satisfactory or not may depend upon a relatively small temperature gradient between the flame and the material heated. Accordingly, preheat of fuel and air becomes critical, and regenerative or recuperative methods must be efficiently applied. Further, suitable port or burner design and the correct disposition of the stream of hot gases in the working chamber may become equally essential.

(iii) Process factors, of particular importance in steel making and metallurgical melting generally, which may introduce the over-riding factors referred to above, are outside the present compass.

In general in hearth and crucible furnaces a fundamental relationship exists between the rate of fuel input and the rate of melting for any particular class of furnace. In Figs. 144 and 145 illustrations are taken from steel making practice in open hearth furnaces. The exact nature of this relationship depends upon a multitude of factors, including the furnace design and that of its auxiliaries as well as the many variables introduced by the process. The determination of this relationship for any furnace investigated is a first step in the study of its fuel efficiency.

The determination of the heat balance can be made according to the methods outlined in the specifications prepared by the British Standards Institution. From the heat balance an indication can be obtained of the directions in which improvements can be made to economise fuel. In hearth furnaces, for example, it can be learnt that apart from heat in the waste gases the greatest loss of heat occurs in the furnace structure, and the deduction is that the solution of the problem lies mainly in the refractories used. Heat losses have been further utilised by the application of the waste heat boiler to the hot gases leaving the furnace system. A typical series of heat balances is shown in Table 91. A line diagram of a typical fixed furnace is shown in Fig. 146.

Modern methods of furnace control involve the use of instruments. Instruments virtually maintain a constant indication of the critical rates of input and of draught and temperature conditions which amount to a watch over those features of the heat balance of most importance in determining efficient operation. The instruments used, in order of importance, are :—

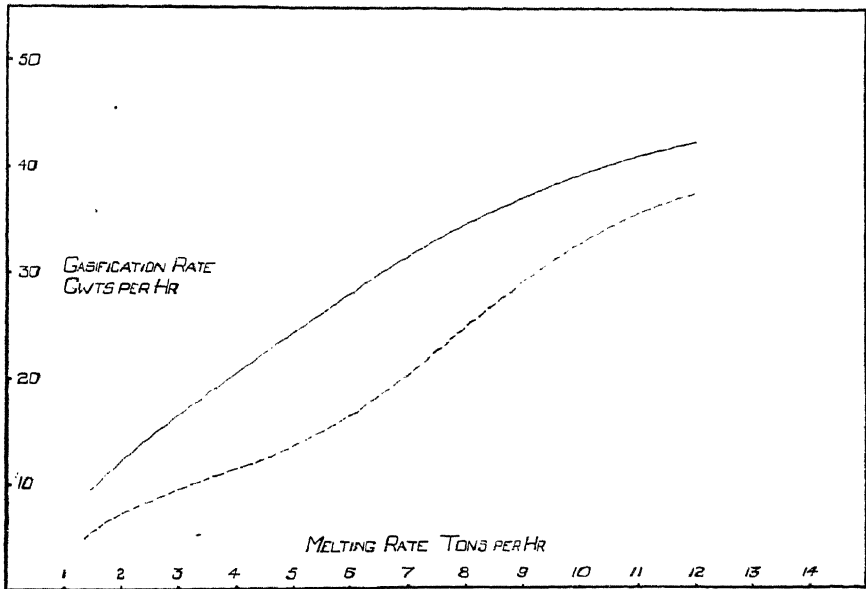


FIG. 144. Relation between rate of fuel consumption and rate of smelting in open hearth furnaces. The band between the curves shows the region in which results have been recorded.

(*J. Iron and Steel Inst.*, Special Report No. 22, Symposium on Steel Making, pp. 248, 539.)

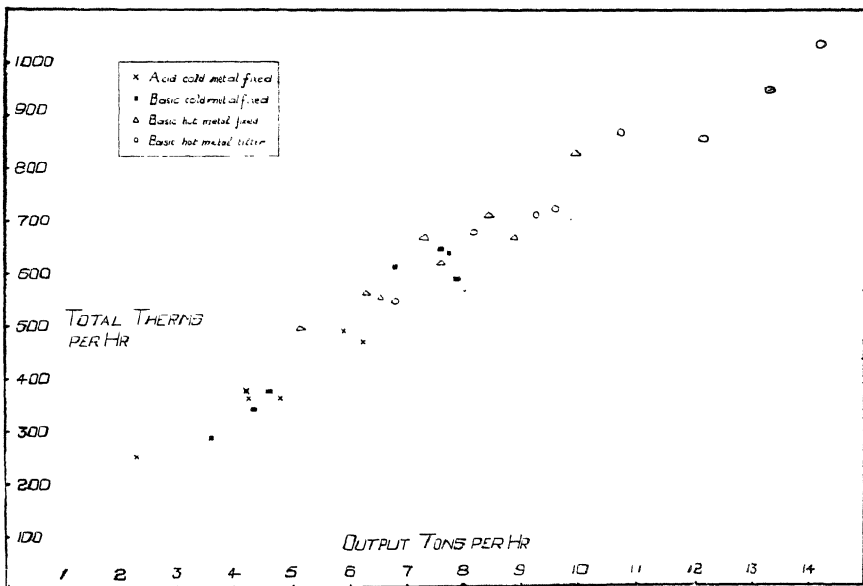


FIG. 145. Relation between the total heat input to the laboratory of open hearth furnaces and rate of output.

(*J. Iron and Steel Inst.*, Special Report No. 22, Symposium on Steel Making, "British Open Hearth Data and their Correlation," O. H. Committee, Iron and Steel Industrial Research Council.)



HEAT BALANCES OF MOLTEN BATH

	5 Basic practice Per cent.	6 Acid practice I Per cent.	7 Acid practice II Per cent.		5 Basic practice Per cent.	6 Acid practice I Per cent.	7 Acid practice II Per cent.
	Thermal efficiency based on heat input at furnace valve Oxidation of metalloids .. Slag formation .. .. Balance from fuel ..	16.1 32.25 4.50 63.25	10.1 57.3 1.2 41.5		8.76 44.1 1.9 54.0	Steel .. .. Slag .. .. Decomposition of limestone Reduction of iron oxides	80.2 15.4 4.4
	100.0	100.0	100.0		100.0	100.0	100.0

REFERENCES: (1) Clements, *J. Iron and Steel Inst.* (1922).

(2) Kinney and McDermott, South Works, Illinois Steel Co.

(3) Combination of gas producers, furnace and waste heat plant, Barnes and Sarjant, *J.I.S.I.*, Special Report No. 22, p. 271.(4) *loc. cit.*, p. 272.

(5) p. 273.

(6), (7) p. 274.

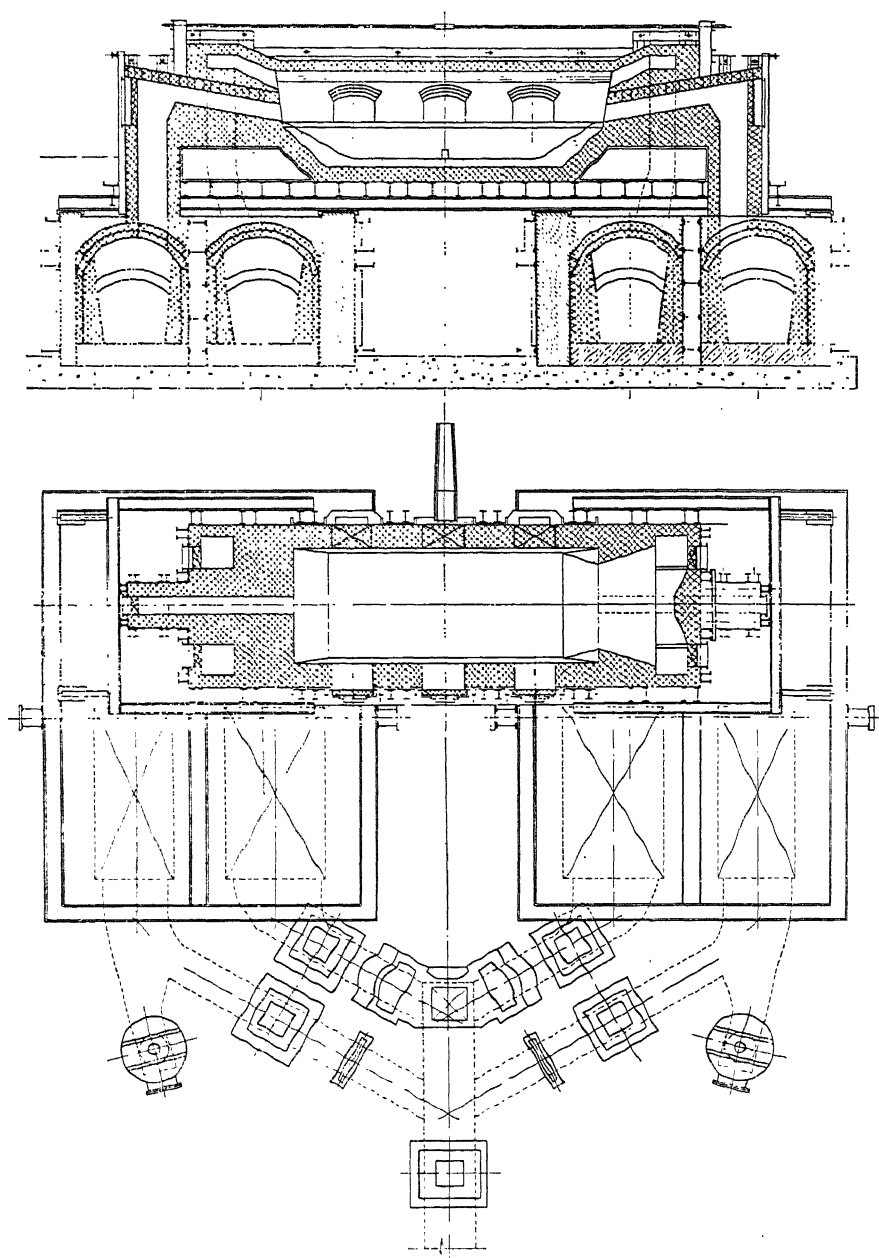


FIG. 146. Open hearth furnace.

(i) Fuel measuring appliances and pressure and draught gauges. These serve to assist the maintenance of the correct rates of firing and conditions of balanced draught necessary to avoid wasteful leakage of gas and air.

(ii) Temperature control of the molten bath or of the furnace lining by means of thermocouples, radiation and optical pyrometers which are of use from the aspect of process as well as fuel efficiency.

(iii) Indicating and recording pyrometers for observing the temperatures in the regenerative or recuperative system. Such instruments not only indicate unsatisfactory temperature conditions in degree, but also in distribution, and ensure the correct timing of the reversals in regenerative practice.

Automatic controls for correct admixture of fuel and air, for correct draughting and for reversal in regenerators, are coming gradually into use as the technical difficulties of applying delicate instruments to heavy industrial processes are being overcome.

### FUEL SAVING IN OPEN HEARTH FURNACES

Notes here given were designed for practical guidance in the operation of hearth furnaces, in particular for steel making, but they have a general bearing on all types of regenerative hearth furnaces.

The problem of saving fuel in open hearth furnace practice is inseparable from problems of steel output and refractory consumption. It does not necessarily follow that using less fuel is saving fuel in the true sense. Eight tons of steel an hour from 40 cwt. of coal an hour may be more economical than 7 tons of steel from 38 cwt. of coal, and is more economical unless the furnace brickwork suffers in consequence. Apart, however, from alterations in furnace design and construction, which are not within the present scope, there are points to be noted in practice which tend to reduce the fuel consumption without adversely affecting output or brickwork and may benefit either or both. A number of these are set out below.

- (1) Gas pressure should be kept at the lowest possible working limit.
- (2) Culverts, valves and furnace brickwork from producers to gas ports should be examined regularly, and repaired when necessary. There should be little or no leakage at these places.
- (3) Combustion in the furnace needs special consideration. Correct mixture of gas and air is essential to give the maximum heat to the bath. If the air and gas are not measured, or the ratio of air to gas not indicated on an instrument, the flame should be closely watched. It should be highly luminous and not too long. The tip should on no account reach the outgoing block. Too long a flame results in damage to the outgoing ports and tends to overheat slag pockets and chequers.
- (4) A slight pressure but not an over-pressure should be kept in the furnace. A slight pressure prevents excessive cold air being drawn in, and saves the amount of fuel necessary to heat the air up to furnace temperature. An over-pressure forces too much flame through openings in the furnace structure, wasting heat outside the furnace and damaging structural steel and brickwork. Normally, a slight flicker of flame visible at the doors is used as an indication of correct pressure. Dampers are the means of controlling the pressure. They should, therefore, be arranged to work easily and should be adjusted according to the amount of gas being burnt. The more gas used the greater the quantity of waste gases formed, and consequently the wider the damper must be opened to give the extra draught required to clear the furnace.
- (5) Gas ports and slopes should be fettled frequently in order to maintain correct alignment and flame direction. The importance of this is often not fully realised, but can easily be proved by trial.



- (6) Reversals should normally be made at regular intervals, although this procedure should be modified if the chequer temperatures have become unbalanced. The object to be aimed at is even heating across the bath and this is much easier to attain if the corresponding chequers at each end of the furnace are approximately equal in temperature. Temperature recorders and controllers installed in the chequer system greatly assist the melter in balancing operations.
- (7) As much information as possible relating to the draught in the furnace system should be obtained. If the pressures and draughts are known at various points when the furnace is going well, the knowledge can often be used to diagnose complaints when the furnace operating condition is unsatisfactory. For this purpose holes should be cut through the brickwork and fitted with easily removable plugs. Readings with a draught gauge may then be taken regularly. Suitable points for the test holes are :—
- (a) In gas and air uptakes—say a foot or so above stage level.
  - (b) In gas and air slag pockets.
  - (c) Above gas and air chequers.
  - (d) In gas and air culverts as near base of chequers as practicable.
  - (e) In the stack flue near the damper and on the furnace side of the damper.
  - (f) Before and after the main gas valve, for obtaining drop in pressure across the valve.

As examples of information obtainable from such readings, if the difference between the pressures before and after the main gas valve is higher than usual, the flow of gas through the valve is being impeded and the valve housing needs inspection and clearing. If the difference between the waste gas draughts taken above and below a set of chequers is greater than usual, the chequers are becoming choked.

- (8) CO<sub>2</sub> recorders are of doubtful value when applied to the waste gases of an open hearth furnace. But analyses should be taken to check correct combustion. If, about an hour after the furnace has been fully charged, Orsat samples are taken through a silica tube inserted into the gas down-take, reliable figures may be obtained. Normal CO<sub>2</sub> content at this point is 15–16 per cent. when operating on producer gas. Care must be taken that no air is drawn into the uptake around the silica tube.
- (9) Instruments to record draughts and pressures continuously are very useful especially when simultaneous readings taken at two or more points are necessary for accurate comparison. But for a permanent installation it is probably better to have quite a small number of recorders, including one showing the gas pressure and another the effective stack draught, rather than a multitude which may confuse rather than inform.

It cannot be stressed too strongly that improvement in fuel consumption of open hearth furnaces depends as much on past as on present knowledge. Data concerning a particular furnace collected when it is going badly may be of little or no value unless they can be compared with similar data obtained when it was comparatively new and efficient. Someone should be responsible for the collection and collation of information. Fuel consumption, steel output, gas pressures and draughting conditions should be observed and set down weekly. This can all be done quite quickly once a routine is established, and the results, arranged as tables, curves or in any other way suitable for easy comparison, will soon be regarded as essential to the attainment of minimum fuel consumption.

## CRUCIBLE FURNACES

Crucible furnaces are fired by coke, gas or liquid fuel, the main consideration in high temperature work being that the fuel must be capable of giving a high intensity of combustion since the quickest melting is usually the most economical from the standpoint of economy of both fuel and metal.

## COKE FIRING

The size and quality of the coke is a factor of major importance. The manner of combustion of coke is related to its physical properties, the harder cokes in a limited supply of air burning with a higher ratio of carbon dioxide to carbon monoxide in the products of combustion, and so giving a higher intensity of combustion. Such cokes of low reactivity are the more suitable for use in high temperature melting operations, such as are carried out in crucibles, reverberatory furnaces and cupolas.

For a discussion of reactivity and combustibility of coke the reader is referred to the literature. It is a subject to which considerable investigation has been devoted. One of the best means of revealing a hot melting coke is the determination of the shatter test.

When coke is burnt on grate bars its ash content, its refractoriness, suitability of grading or size analysis, the draught available and the critical rate of combustion required to give quick melting must all be considered in relation to the work in hand. In the older type of natural draught crucible furnace or pothole, fired by coke, due to limitations of design the old beehive coke gave the most satisfactory and economical melting at the highest temperatures, but with adequate draught and a suitable size of coke, by-product oven coke is equally effective, provided a clean coke of low reactivity and good shatter resistance is used. The sulphur content of the fuel must be limited.

The quantity of coke charged at each feed and the draught must be adjusted to follow the course of the heat, so that fresh charges are not required just as the heat is nearly ready.

A suitable specification by way of guidance on general principles only for a coke of good melting quality is as follows:—

Moisture.	As low as possible.
Ash.	8 per cent.
Sulphur.	Under 1 per cent. (Purity and good shatter index permits a higher sulphur content to be carried.)
Shatter index.	1½ inch index as high as practicable and preferably over 90 per cent.
Size.	Less than 6 per cent. through 2 inches, the remainder being as close graded as practicable for the particular size desired.

## GAS AND OIL FIRING

With oil or gas-fired crucible furnaces the use of the correct rate of fuel consumption and fuel-air ratio is of prime importance. Fuel is wasted if too much or too little fuel or air is used. The optimum conditions are indicated from the shape and colour of the exhaust flame, and gauges indicating flow, pressure and draught are called for to assist control. A burner out of line will readily cause waste of fuel, as well as spoil metal.

The size of crucible, and shape and dimensions of the combustion chamber and off-take flues are primarily a matter of initial design, but in many operations are amenable to alteration at rebuilds. Preheater tops, made from an old crucible are sometimes employed with the object of getting the whole of the charge into the furnace at the start of the heat. Worn linings should be constantly kept in repair, carbon deposits prevented from forming in combustion chambers and all leakages avoided.

### SHAFT FURNACES

The most important shaft furnace is the blast furnace for the smelting of metals from their ores, but since this type of furnace is of specialised type, and its operation is bound up closely with a complicated metallurgical operation, it is not considered in a volume intended for general study.

Another important type of shaft furnace for melting purposes is the cupola furnace.

### THE CUPOLA FURNACE AND FUEL ECONOMY

The cupola, when compared with other coke-fired melting furnaces, has a

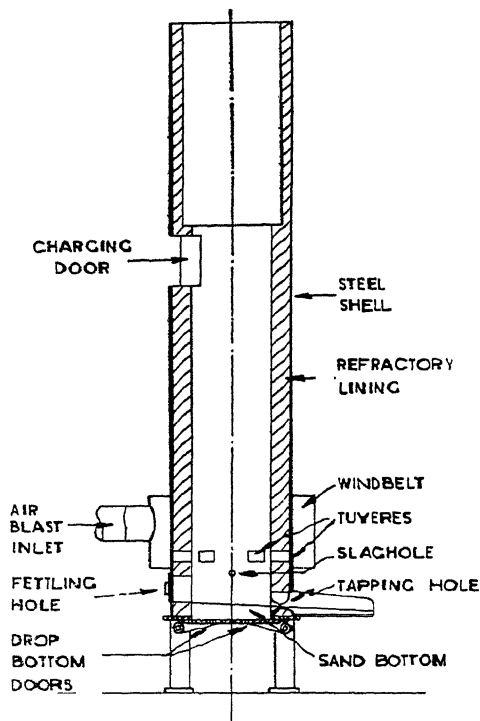


FIG. 147. Drop bottom cupola.

relatively high thermal efficiency and because it will give reasonably satisfactory results even when inefficiently operated, it is sometimes found that little attention has been paid to design and operation. It is intended here to show how design and operation can affect the thermal efficiency of cupolas and consequently their fuel consumption.

### DESIGN

The cupola (Fig. 147) consists of a mild steel shell erected in a vertical position and lined with refractory. The shell is mounted either on a brick foundation or on steel columns. The steel column arrangement is used on most modern cupolas and the bottom of the shell is provided with drop-bottom doors, through which the debris, consisting of coke, slag, etc., can be discharged at the end of a melt.

In drop-bottom cupolas, the working bottom is made up with moulding sand which covers the drop doors. This bottom slopes towards the metal tap-hole at the front of the cupola, the tap-hole being at the lowest point. The sand bottom is put in through a hole in the shell at the back of the cupola opposite the metal tap-hole. This is known as the fettling hole or cleaning-out door, and in solid brick-bottom cupolas is also used for raking out the debris at the end of the melt.

Tuyeres for supplying the air blast are provided at a height of between 2 and 4 feet above the working bottom. The air is delivered to the tuyeres from a wind belt, which surrounds the shell. A high pressure fan, or a blower of the Roots type, supplies the air to the wind belt through a blast pipe. A valve is provided in the blast pipe so that the air supply may be controlled.

A slag hole is located at a level about 9 inches below the centres of the tuyeres. A charging hole is provided through which metal, coke and flux are fed into the furnace, and this is situated 10 to 20 feet above the tuyeres, according to the size of the furnace. The shell is usually continued for 15–20 feet above the charging hole to form a chimney.

## OPERATION

In preparing the cupola for a heat, a fire is lighted on the working bottom and a bed of coke built up to a level just above the tuyeres. This coke bed must be thoroughly hot before it is finished off to its final height. The fire may be drawn up by natural draught, or a forced draught may be used, supplied by a small auxiliary blower or by means of kindling tuyeres from the wind belt. As soon as the fire is well alight, the fettling hole is closed up with sand and a cover plate fixed over it. The coke at the tuyeres must be well heated before charging begins. It is advantageous to finish off the bed with about 12 inches of cold coke to bring it to its correct level. This level should be checked by means of a measuring stick inserted at the charging hole. When the bed is completed, the cupola should be charged full to the sill of the charging hole before the blast is put on. Alternate charges of metal, coke and flux are used until the cupola is full. When the metal in the cupola starts melting, the rate of charging should be equal to the rate of melting, so that the furnace is kept full throughout the heat.

The metal should begin melting in five or six minutes after the blast is put on. The molten metal may be collected in the well of the furnace below the tuyeres and tapped off as required, or allowed to run from the furnace as it melts and collected in a ladle or receiver. When slag accumulates in the well, the slag hole is opened and the slag run off, preferably into a bogie for easy removal.

At the end of the melt, charging is stopped and the blast is kept on until all the metal has melted. A careful watch should be kept at the tuyeres and the blast should be shut off immediately it is seen that no more metal is melting. If this is not done, coke which could otherwise be recovered is wasted. By far the greater part of the damage to the lining takes place during this emptying or "blowing down" period, and unless the blast is shut off as soon as all the metal is melted, the wear on the lining will be increased.

As soon as the blast is shut off, the debris should be discharged from the cupola, either by dropping the bottom doors, or, in solid-bottom cupolas, by removing the fettling hole cover and raking out. The debris should be immediately quenched with water, using an ample supply. Care should be taken to see that it is completely black before the water is turned off. It is frequently found that the draught through the cupola will draw up any remaining embers and re-kindle the coke, which will then burn away during the night. When the debris is cold, usually the next morning, it should be hand picked and the

coke remaining returned for use in the cupola. This coke has not suffered in any way by being used in the furnace, tests having shown that it is not contaminated by any deleterious elements and is quite suitable to replace new coke on the charges.

## TUYERES

The total area of the tuyeres should be one-fourth to one-seventh of the cross-sectional area of the cupola inside the lining at tuyere level. The shape of the tuyeres, whether round, rectangular, flared or parallel, has little influence on the efficiency of the furnace, so long as an adequate area is provided inside the lining. Some cupolas are provided with tuyeres in one row; others have two or more rows. In the balanced blast and Pournay types, using multiple rows of tuyeres, the tuyeres in the upper rows are provided to admit air for the more complete combustion of the coke to carbon dioxide. In other types, the use of a double row of tuyeres means only that an additional quantity of coke is required for the bed each time the cupola is used. Unless a specially high pick-up of carbon is required, necessitating the use of a deep bed, this is wasteful and increases the fuel consumption of the furnace.

For the melting of high steel charges for the production of steel by the converter process, for example, a high carbon content is not required, and tuyeres arranged in a single row will give the best results.

## INTERNAL DIAMETER AND HEIGHT

The internal diameter of the furnace at the tuyeres and in the melting zone controls the output obtainable from the cupola. This assumes, of course, that the air supply is adequate for the operating diameter. In normally operated cupolas, an output of 0.7–0.8 ton per hour is obtained per square foot of cross-sectional area in the melting zone. When melting high steel charges, however, the rate of melting is reduced, and it is also desirable to operate the furnace more slowly than is possible on pig and scrap-iron charges. The output when melting charges containing 80–100 per cent. steel is seldom more than 0.5 ton per square foot cross-sectional area per hour.

The shape of the lining is important and is governed by several factors. In cupolas which are tall between tuyeres and charging hole, a parallel lining from base plate to charging hole may be used, and this also holds for cupolas where a low output is required for the internal diameter. In short furnaces, and where a high rate of melting is required, a boshed lining is desirable. The holding capacity of the furnace for solid metal, coke and flux to a large extent controls the temperature at which the exhaust gases are discharged. The smaller the amount of stock held in the cupola, the hotter the exhaust gases and the greater the loss of heat from the fuel used. Except in very small cupolas, i.e. less than 24 inches internal diameter, where there is a danger that the stock will bridge and hang up in the furnace, the preheating zone above the coke bed should hold metal, coke and flux sufficient for one hour's melting.

Many cupolas are installed which are short between tuyeres and charging hole and owing to the short preheating space, the gases escape at a very high temperature and the thermal efficiency is relatively low. Conditions in these designs can be considerably improved by increasing the internal diameter from a level near the top of the bed coke and maintaining this increased diameter up to the charging hole. The thickness of the brickwork in the expanded portion must not be reduced below, say, 6 inches on cupolas of 5 tons capacity, or 9 inches on large cupolas. The area in the expanded portion may be 30–40 per cent. greater than in the tuyere zone. The flare from the smaller to the larger diameter should be gradual and is usually made about 2 feet 6 inches deep.

Because of the loss of heat when the stock column is short, it is important that the cupola shall be kept charged full throughout a melt.

A boshed lining makes it a little more difficult to separate different mixtures when irons of a number of different compositions are melted in one heat. It is not desirable, also, when a cupola is completely emptied and re-charged several times a day. Because of the additional bed coke required and the severe wear caused at the lining, this latter method of operation should be avoided whenever possible.

An alternative to increasing the internal diameter in the preheating zone is to increase the height of the charging hole, but this usually necessitates major alterations to the charging platform and hoisting equipment. However, in some cupolas, operating for long periods, which are very short to the charging hole, the alterations necessary to enable the furnace to accommodate one hour's supply of metal in the preheating zone are well worth consideration.

### REPAIRS TO LINING

During every melt in a cupola, part of the lining, especially that in the zone of maximum temperature immediately above the tuyeres, is slagged and eroded away. This material must be replaced after each melt by patching with ganister and/or firebricks, ramming round a former, etc. Whichever method of repair is adopted, the furnace should be brought back to the same internal dimensions each day. Slipshod repair work causes increased fuel consumption as well as short life of the lining and troubles in operation. If the repair work is poor and insufficiently dried out before the cupola is put to work, the new material will rapidly come away and heat must be consumed in converting it into slag.

A more important point is that if the internal dimensions are allowed gradually to become larger, the amount of coke used on the bed is increased. The efficiency of the furnace may also suffer, thus necessitating the use of more coke to produce metal of a suitable temperature. Whether the furnace is being patched to its correct dimensions can be found by occasionally checking the weight of coke required to bring the bed to its correct height.

### HEIGHT OF COKE BED

The height to which the bed coke is built up above the tuyeres is an important factor in controlling the temperature of the metal from the first tap and also from subsequent taps. Other things being equal, a low bed will yield cooler metal than one which is high. A bed height of 2 feet above the tuyeres, that is, above the top row of tuyeres if two rows are used, is somewhere near the minimum height which may be employed. Where extremely hot metal is required, or where a high pick-up of carbon is desirable, a bed height of 4 feet may be necessary. In exceptional cases, as for example in producing metal for whiteheart malleable castings, an even higher bed may be required and it is, of course, false economy to jeopardise the temperature and quality of the metal by starting with a bed which is too low.

### METAL AND COKE CHARGES

The weight of the metal charges has some influence on the efficiency of the furnace. A large number of light charges will, under certain conditions, give a lower thermal efficiency than a smaller number of heavy charges. No hard and fast rule can, however, be laid down regarding the size of charges for various sizes of cupolas as this depends largely on the type and quality of the metal being melted. The heaviest charge a cupola will melt successfully is about one-sixth of its hourly output, but it is often advisable to use charges one-tenth or less of the hourly output.

All metal charges should be weighed. The coke charges should also be weighed, but it is sometimes difficult to provide the necessary labour and other facilities. The coke should then be measured in a skip provided for the purpose. If the skip is of a suitable size, that is, just holding the correct weight of coke for a charge, this method of measuring will be found almost as accurate as weighing. Careful control of the weight of metal and coke is well worth while as it not only invariably results in a saving of fuel, but is also often found to yield an increased output and metal of a more uniform temperature throughout the heat. Charging coke by shovels or forksful, on the other hand, is unsatisfactory and usually leads either to a waste of fuel or to fluctuating metal temperature. The flux, usually limestone, should be approximately 25 per cent. by weight of the coke charge and may be weighed or measured in the same way as the coke.

### COKE RATIOS

If all metal charges are weighed and the coke either weighed or measured accurately by volume, it is possible to assess the ratio between metal melted and coke charged. This ratio depends on a great number of factors, including the design of the furnace, amount of preheating space available and so forth; the type of metal to be melted, size of individual pieces, amount of steel scrap included in the charge, etc.; the quality of the metal, e.g. whether high or low phosphorus, or whether a high carbon pick-up is required. Generally speaking, the smaller the size of the pieces of metal charged, the lower the coke consumption in melting them. Large pieces should therefore be broken as far as this is possible. The higher the percentage of steel melted, the more coke is required to melt and carburise it.

Very small pieces of steel, such as punchings, oxidise very easily in the cupola and necessitate the use of additional coke if they are to be melted successfully. Owing to the difficulty of accomplishing this, even under the best conditions, their use should be avoided. Very heavy steel scrap requires a high coke consumption to ensure that pieces are completely melted before they descend to the oxidising zone near the tuyeres. If a high carbon pick-up is required, the coke consumption will generally be high, as carbon must then be supplied to be taken up by the metal. Cupolas operated intermittently with frequent shut-downs will require more coke than those run continuously.

Owing to the large number of factors involved, it is not possible to give definite recommendations for the ratio of metal melted to coke charged which can be achieved on different classes of work. The following table is intended only as a guide and shows good average practice in the industry.

Class of work	Metal to charge coke ratio
High-phosphoric iron for fairly heavy castings ..	12 1-15
High-phosphoric iron for light castings .. ..	11 1-14
Medium-phosphorus iron for engineering castings	10 1-12
Low-phosphorus iron for high duty and automobile castings .. .. .	1-10
Charges containing 50-75 per cent. steel scrap ..	1- 9
Charges containing more than 75 per cent. steel scrap .. .. .	6:1- 8

In examining the overall coke consumption in a cupola, the coke used on the bed must be taken into consideration. The coke recovered from the debris at the end of the melt should be deducted from the total weight of bed coke used. As the same amount of bed coke is required whether the furnace melts 5 or 50 charges, it will be seen that the overall coke consumption depends to

some extent on the length of the heat, and coke consumption per ton of metal melted will be less on long than on short heats. In very small foundries, therefore, where it can be arranged, it is better from the fuel economy point of view to have a long heat every other day than a very short heat each day.

In conclusion, nothing is so conducive to fuel economy and overall economy in foundries as the maintenance of a supply of metallurgical coke of high and uniform quality. If the quality fails, the consumption is increased, with its consequence of increased cost of operation and transport; the metal quality and temperature suffer as a result of the increased sulphur and ash content of inferior fuels. Foundry coke is not only a fuel, but a technical material, the quality of which affects the quality of the metal melted. Hence fuel economy in the narrow sense, may from a wider point of view actually militate against economy and quality of product. All fuel economy proposals involving melting coke must be judged on this basis.

## 3

**SINTERING FURNACES**

The cement kiln may be regarded as the most typical example of the application of heat for sintering. It concerns also an important industry. The following statement outlines the methods of operation and maintaining efficiency in rotary cement kilns.

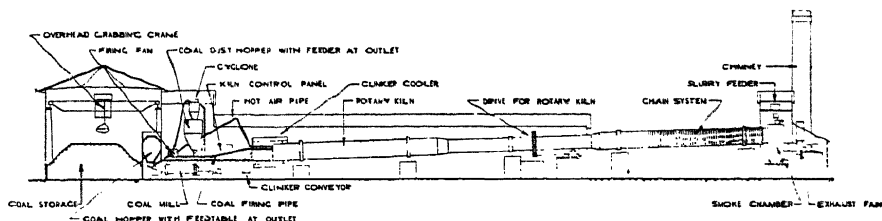


FIG. 148. Rotary cement kiln fired by pulverised coal.

**ROTARY CEMENT KILNS**

Rotary cement kilns as used in the cement industry, consist of cylindrical furnaces supported on rollers (Fig. 148). They vary in length from about 150 feet to something like 500 feet, and in diameter from 7 feet to 12 feet.

Nearly all cement works in this country are working on what is called the "wet process," which means that the raw materials are fed into one end of the kiln as a wet slurry and, owing to the rotation and inclination of the kiln, gradually gravitate towards the other end, where they emerge as clinker.

Pulverised coal is used in cement kilns as the fuel. Fuel oil or natural gas are also used sometimes in other countries.

The raw materials fed into the upper end of the kiln may contain between 36 and 43 per cent. of moisture. When entering the kiln, the slurry meets the hot combustion gases, the water is evaporated and later, when the material reaches a temperature of 1,650° F. (900° C.), the carbon dioxide is driven off; finally, the material is heated to a temperature of about 2,650° F. (1,454° C.) in the hottest part of the kiln where the chemical reactions take place which cause the formation of the hydraulic product—cement clinker (Chapter XXIX.).

The clinker is generally cooled through a clinker cooler which either forms part of the kiln or is a separate unit. It is cooled by means of an air current and, therefore, a substantial part of the latent heat contained in the clinker is recuperated and utilised for preheating the secondary air required for burning the coal in the rotary kiln.



In order to produce the necessary chemical reaction and thereby give the required strength of the cement, it is essential to reach the above-mentioned temperature of about 2,650° F. (1,454° C.). This temperature is close to the fusion point of the refractories, and were it not that it is possible to build up a coating of cement clinker on the refractories, it would be difficult to get any lining material to withstand the heat and the abrasion.

Provided complete combustion is obtained with a minimum of excess air, and the moisture content of the slurry is kept as low as possible, fuel saving can only be looked for in the reduction of the temperature of the exit gases and of the cooled clinker.

To reduce the temperature of the exit gases, cement kilns are generally provided with a drying zone, which may have chains or other devices for the purpose of increasing the area opposed to the gases and external driers are also sometimes utilised. There is little which the operator can do to effect a saving of fuel in this part of the kiln other than to see that chains and other drying elements are kept in as efficient a state as possible. This also applies to the recovery of heat in the cooling of the clinker.

The coal consumption in rotary cement kilns operating on the wet process may vary from about 24 per cent. to about 32 per cent. when based on standard coal having a calorific value of 12,500 B.Th.U. per lb. The variation may be due to different factors, such as varying moisture content of the slurry, nature of the raw materials and the efficiency of the kilns. To make any improvement in the kiln efficiency would mean a considerable amount of reconstruction which it would be difficult to undertake in present conditions.

Thus, apart from keeping the mechanical parts of rotary cement kilns in good repair, saving in fuel can only be effected by keeping the moisture content of the slurry as low as possible and by obtaining as complete combustion as possible.

Most cement works use a gas coal having a volatile content of about 25–30 per cent. The coal should have as low a moisture content as possible, depending on the facilities for drying and grinding it and, when ground, the fineness should be about 15–20 per cent. residue on a 170-mesh sieve. Ash content should preferably be as low as possible, but it should always be of such a nature that it will assist in forming a protective coating on the refractory lining.

If coal having too high an ash content, or ash of an unsuitable nature, is used, it frequently leads to the formation of clinker rings, which not only tend to reduce the output of the rotary kilns, but also lead to inefficient combustion and incomplete burning of the clinker.

To ensure complete combustion, cement manufacturers generally reckon that it is advisable to have about 1 per cent. of oxygen in the exit gases. Frequent analysis of the exit gases should be made, and if an oxygen recorder is not already installed, it will be found a valuable instrument for the purpose of checking the combustion. It is very important to ascertain that samples of the exit gases are truly representative and, in this connection, it may be worth while noting that experience has shown that the most representative sample is extracted in the upper left-hand quarter (10.30 on the small hand of a watch) when looking at the kiln from the back and assuming it rotates in an anti-clockwise direction. Where two or more coals of widely varying qualities are used, it is essential for them to be well mixed so as to minimise variations in volatile contents, ash, and so forth; otherwise unstable conditions are set up in the kiln, causing incomplete combustion. Provided really good mixing facilities are available, it is possible for cement works to use a proportion of anthracite having a volatile content as low as 6–8 per cent.

## KILNS AND OVENS

Kilns and ovens are the structures used for the heating or firing of a number of important products, particularly those made by the pottery and heavy clay industries from clay or a mineral base. Differences in the type of kiln, together with great differences in the thermal requirements of the raw materials used to obtain the properties desired in the fired ware involve perhaps a greater number of problems specific to the process used than are associated with any other type of heating appliance. This is accentuated by the fact that the production of high yields of first quality ware is usually of major importance. Specifications for the testing of kilns used in the heavy clay industries have been prepared by the British Standards Institution (B.S.S. No. 1081, November, 1942, "Test Codes for Kilns for Heavy Clay Ware, Including Refractory Materials"). Similar specifications are being prepared for kilns used in the pottery industry. Reference to these standards will indicate those features which have to be watched in order to maintain or improve thermal efficiency. It is not proposed to deal here with the efficiency of such plant in a detailed way, but mainly to indicate certain outstanding factors affecting fueleconomy.\*

In the firing of all clay wares the need is recognised: for heating to a specific time-temperature schedule determined by the thermal requirements of the material; for obtaining uniform heating and control of the process involved; and for maintaining correct control over kiln atmosphere according to whether oxidising or reducing conditions are required to produce the desired changes in the ware.

Three stages of:

- (1) After smoking period in which appreciable volumes of hot air must be passed through the setting in order to remove moisture until the ware is completely dry.
- (2) The oxidation or preheating period in which chemically combined water is removed and oxidation of any carbonaceous matter in the green product is safely completed.
- (3) The finishing period during which the required final temperature is attained and soaking time allowed at the finishing temperature for obtaining uniformity of heat treatment and developing the degree of vitrification or maturing properties required. Distinctive methods of pyrometry are used to determine when correct finishing conditions have been reached, e.g. seger cones or other heat recorders used to measure the heat work performed.

In each of these stages, the combustion conditions used are governed by special requirements for different classes of goods and the firing technique required usually differs markedly from that used in other types of heating plant. Firing methods also depend on whether intermittent or continuous kilns are used and on the applicability or availability of automatic firing mechanism. Intermittent kilns are mainly used to fire special products not amenable to continuous practice or to secure flexible treatment on a works manufacturing a variety of products where flexibility is of more importance than high thermal efficiency or large output of any one product. Unavoidable heat loss from the firing of these kilns is considerable. Continuous kilns are specially applicable to the firing of standard products where large throughput is desired. Recuperation of heat from cooling goods and from the kiln gases makes firing in this kind of kiln an operation of high potential thermal effi-

\* The reader is referred to two publications by the British Pottery Research Association, viz.: (1) "Collected Talks Given at an Intensive Refresher Course in Fuel"; (2) "Fuel Saving in the Refractories and Heavy Clay Industries: Part I.—The Firing of Intermittent Kilns; Part II.—The Firing of Continuous Kilns of the Belgian Type."

ciency. Kilns of both classes are used in the heavy clay and in the pottery industries, but there are often differences in the types used in each industry. In general, coal firing is used on intermittent kilns, but graded or ungraded coal, pulverised coal, town gas, producer gas, oil, or electricity may be used for firing continuous kilns according to the quality and class of product made.

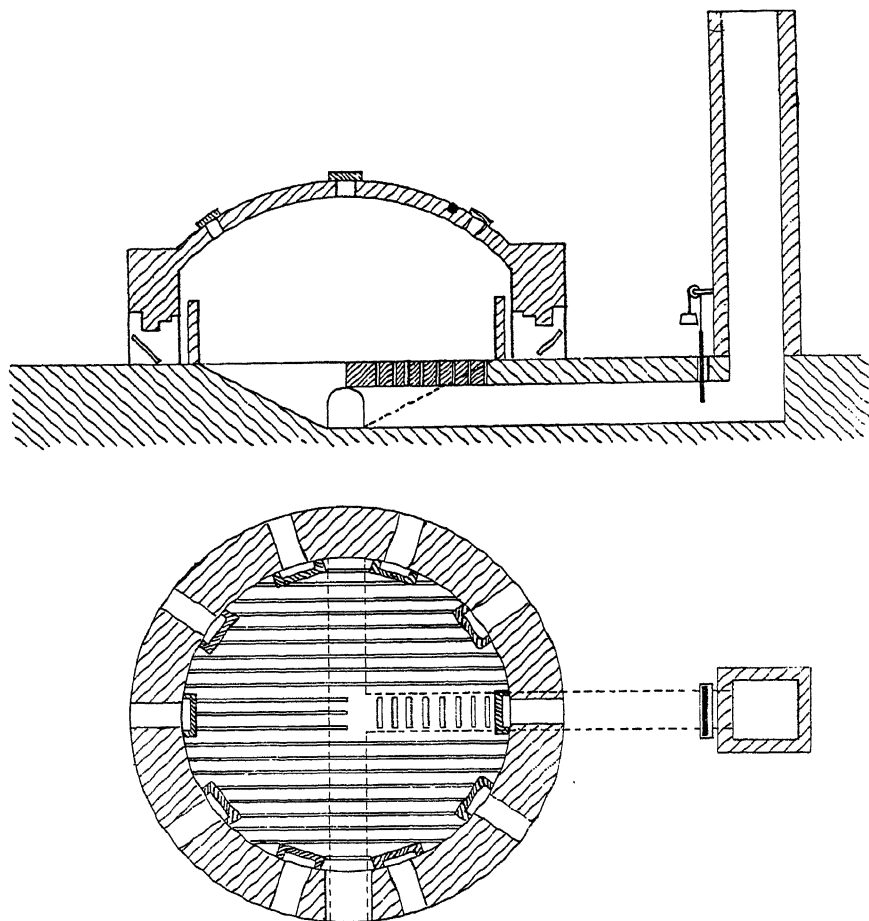


FIG. 149. Down-draught kiln for firing heavy clay ware.

In this country the use of mechanical stokers is largely confined to continuous kiln practice.

#### TYPES OF KILN AND OVEN

(a) *Intermittent Kilns.* Two main types of intermittent kiln are used in the heavy clay industry: the rectangular down-draught and the round down-draught. Both muffle and open-flame conditions are used with each type. In muffle firing the gases from the fires are not allowed to make contact with the goods being fired, heat transfer to the goods being obtained almost entirely

by radiation from the muffle walls. With open-flame firing, which is used to a much larger extent, all gases and flames from the fires pass through setting spaces among the ware before the combustion products are finally exhausted through the flue system. Apart from the shape and type differences generally recognised, intermittent kilns vary widely in size and in the arrangement or disposition of the sole flue system.

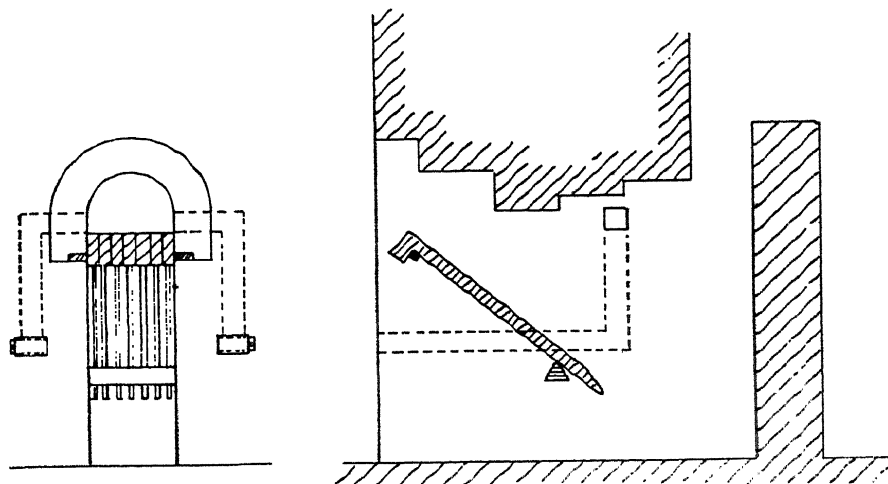


FIG. 150. Firemouth provided with inclined grate bars as used in the heavy clay industry.

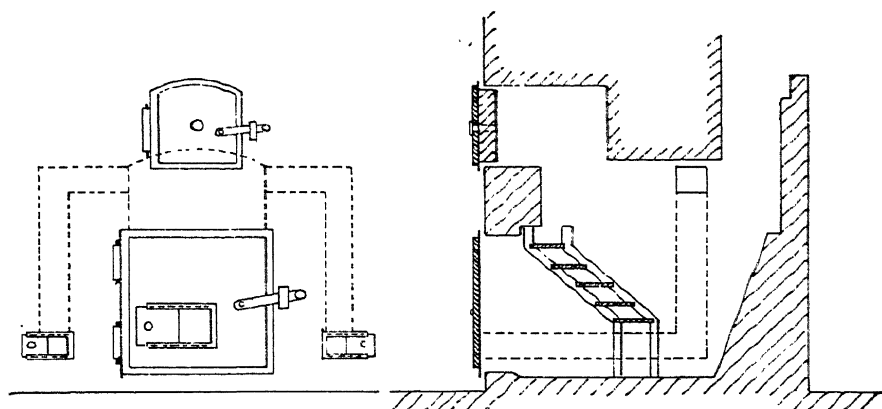


FIG. 151. Firemouth, stepped grate type, provided with top and bottom firing doors.

Each kiln is usually operated from a separate stack, generally external to the kiln, though a central internal stack is used sometimes. Draught on the kiln is controlled by means of a damper at the base of the stack. The number and disposition of the firemouths also varies. These may be of the solid bottom type or they may be equipped with horizontal, inclined or step grates. Combination grates are also used. Modern kilns are usually fitted with fire-doors permitting control of primary air, and slide-controlled secondary air flues

are being increasingly used. The designs of a typical round down-draught kiln and firemouths are shown in Figs. 149, 150 and 151.

In pottery practice two types of kiln or oven are distinguished—the up-draught and the down-draught. Each of these types may be fired on the muffle or open-flame system. Kilns are to be found in a wide variety of forms suited to the special needs of the product fired. The disposition of the firemouths plays an important part in obtaining correct temperature distribution.

In the up-draught type of kiln, an illustration of which is given in Fig. 152, the products of combustion are evacuated through openings in the crown of the kiln. These may be surmounted by a superstructure, termed the cone, which augments the draught for operating the kiln and exhausts the gases from the

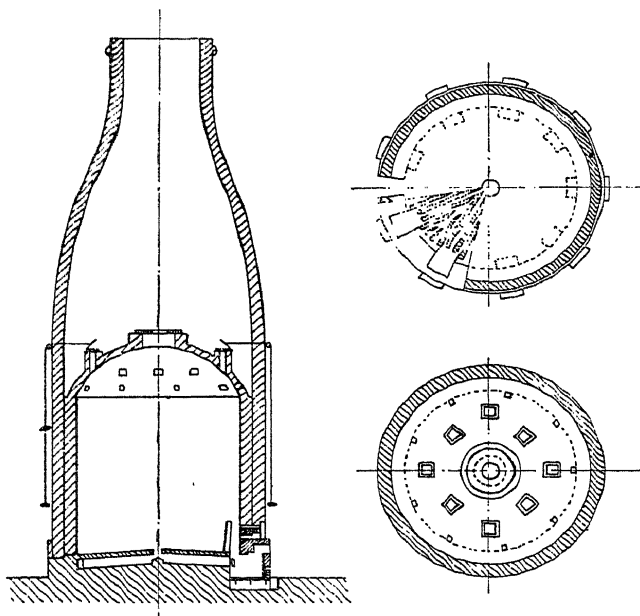


FIG. 152. Up-draught pottery kiln.

various outlets. In this type of kiln a certain amount of draught is naturally provided from the stack effect of the kiln itself. Control of the firing is effected by the adjustment of check dampers placed in the outlets in the crown or other parts of the kiln. The disposition and size of kiln outlets varies appreciably.

In down-draught pottery kilns the gases from the firemouths pass over or through the ware before they are withdrawn through openings in the floor from which they travel through vertical flues between the firemouths into the cone.

In the Wilkinson type of kiln part of the gases are exhausted through openings in the crown and part through openings in the floor. This arrangement involves a combination of up-draught and down-draught conditions. Muffle kilns used for pottery firing do not differ greatly in principle from those used in the heavy clay industry, but the cone system is generally used for exhausting the waste gases, and the kilns are generally of comparatively small size.

<sup>a</sup> (b) *Continuous Kilns.* The economic advantages of mass production and high thermal efficiency are obtained from the use of continuous kilns (Fig. 153).

Because of the low fuel consumption and the refinement in control offered, advantages are sometimes obtained by firing with fuels other than coal.

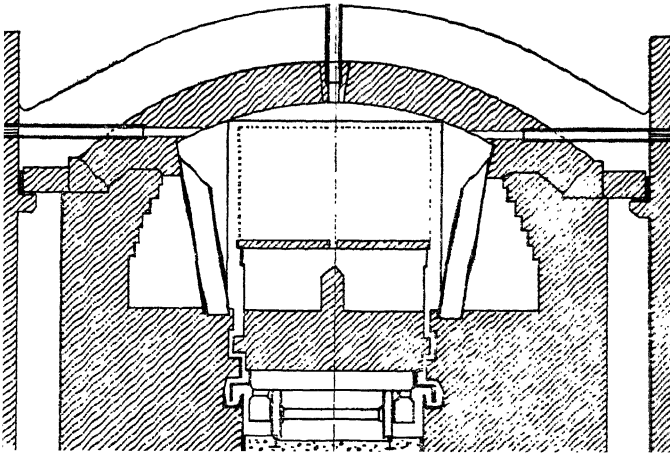


FIG. 153. Continuous kiln : (A) Thrift.

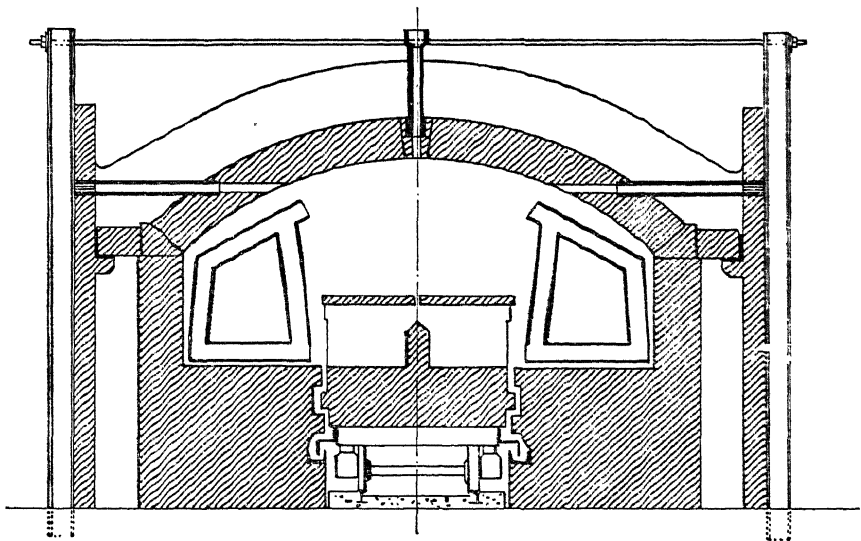


FIG. 153. Continuous kiln: (B) Dressler.

Two distinct firing principles are used in continuous practice. In car tunnel kilns the preheating, firing and cooling zones are fixed and the goods travel through these zones on cars or bogie carriages operated by an external pusher

mechanism. These tunnels may be either straight or annular, a moving hearth being used in the annular kiln instead of cars. The straight tunnel has the advantage of acting as a conveyor between factory departments and economy of labour may be combined with the efficient use of fuel. Such plant assists in maintaining a flow of material on modern mass production lines.

In the second type of continuous practice the goods are set in the kiln and remain stationary while preheating, firing and cooling zones move round the kiln. With this type, one continuous tunnel may be used as in the Belgian and Hoffmann kilns or the kiln system may consist of a number of transverse arch

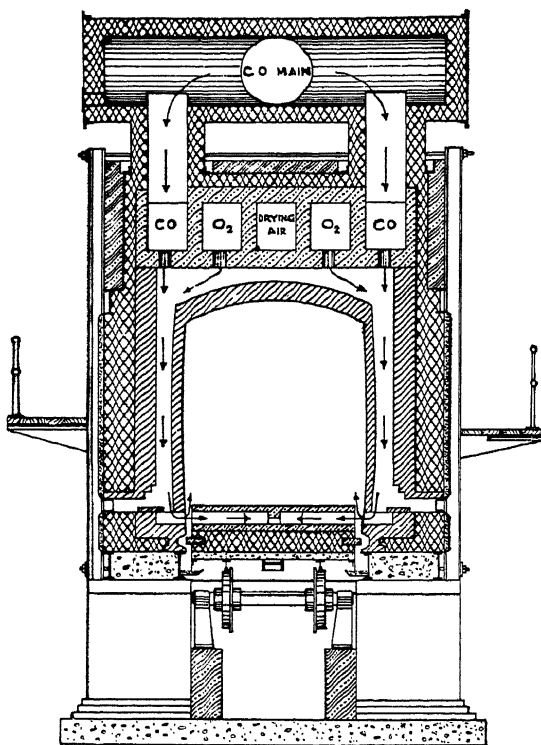


FIG. 153. Continuous kiln: (C) W-D circular tunnel kiln.

chambers connected through suitable chamber openings. When the setting is stationary the kilns may be built to include a number of independent firing zones. Among the advantages of this system are compactness of layout and, in transverse arch chamber kilns, chambers of very large capacity may be used. Semi-continuous kilns are also used to obtain certain of the advantages of continuous practice.

With all continuous kilns heat from cooling goods is used to preheat secondary air required for combustion and primary air may also be preheated in this way. Some of the heat recovered may be transferred to different parts of the kiln when more or less complicated systems of hot air circulation are used, or hot air may be transferred for extraneous use in dryers. The hot kiln gases are used to preheat the incoming or cooler ware so that the main heat losses

are confined to those due to radiation, convection and conduction from the outside walls.

In the heavy clay industry, straight car tunnel kilns, ring tunnel or annular kilns, such as the Hoffmann and Belgian, transverse arch chamber kilns, and semi-continuous kilns are used. Car tunnel kilns and Belgian kilns are mainly used in the refractories industry whereas Hoffmann and transverse arch chamber kilns are very widely used for firing building bricks. Open-flame methods of firing are generally used for heavy clay wares. Car tunnel kilns are invariably fired from the side. Mechanical stokers are usually used for coal firing, but these kilns may also be fired by producer gas. Belgian kilns are also side fired, the grate then extending across the width of each chamber and hand firing is generally adopted. The Hoffmann and transverse arch chamber kilns are top fired with or without the use of fenders or grates. The coal is usually fed to charging shafts left in the setting through a series of small holes in the crown about 4 feet apart. Certain transverse arch chamber kilns are side fired and firing by producer gas is also used for better quality ware.

Continuous kilns used in the pottery industry are mainly of the straight car tunnel type or annular type with moving hearth. Open-flame, muffle, or semi-muffle conditions of firing may be used. These kilns are generally fired with town gas or producer gas, but electrical heating is employed to some extent.

In addition to the above classification, kilns in the pottery industry are also classified according to the purpose for which they are used, e.g. for firing biscuit, glost or enamel ware.

## FUELS

The type of fuel depends on the kiln used and the material fired. General requirements are shown in Table 92.

TABLE 92. KILNS AND FUELS

Kilns	Fuel
<i>Heavy clay—</i>	
Intermittent down-draught (open-flame, muffle and semi-muffle).	Coal: fired to side mouths.
Semi-continuous kilns .. .. .	Coal: fired to side mouths or top fired; fenders or grates may be used.
Continuous car tunnel kilns .. .. .	Coal: mechanically fired; producer gas; pulverised coal and fuel oil also used.
Continuous transverse arch chamber kilns	Coal: top fired; fenders or grates may be used. Producer gas used for special products.
Continuous ring tunnel or annular kilns ..	Coal: Hoffmann top fired; Belgian side fired on to grates.
<i>Pottery—</i>	
Intermittent, up-draught, down-draught, up- and down-draught, open-flame and muffle.	Coal: fired to side mouths.
Continuous car tunnel or annular moving hearth.	Town gas, producer gas, electricity.

Coal is the most usual fuel for kiln firing. For most purposes, size and size grading are important. Small coal is preferred for top firing and usually for use on mechanical stokers. For stoker firing, graded coal of small size is sometimes required. For intermittent kilns larger size or size grading is often specified, although "run of mine" coal is used if the proportion of fines is not too high. Correct size grading is of special importance during the finishing



period. Close grading is usually required for Belgian and fender fired kilns. Special grates may be used to suit the type of coal available.

Long flame coals with volatile matter content above 32 per cent. are required since radiation from long flames has considerable bearing on the uniformity of heating. For high thermal efficiency the ash content of the fuel should be as low as practicable. Low ash coals are specially desirable for firing to fenders and grates of the Belgian type where the ash may not be removed during the firing. Low ash content becomes more important with increased finishing temperature. Top fired kilns, such as Hoffmann and transverse arch chamber kilns, can usually deal with coals of higher ash content especially if the firing temperature does not exceed  $1,000^{\circ}\text{C}$ . Swelling or strongly coking coals are not generally desirable because they restrict the flow of primary air through the fuel bed. Where small coal is used for maintaining a thick fuel bed, however, some use may be made of coking properties if the bed is poked with sufficient frequency. Fusible ash is detrimental in general practice. Coals of low sulphur content are usually required for the firing of most pottery ware.

## FIRING :

### (1) INTERMITTENT KILNS AND OVENS

The principles of combustion as outlined in earlier chapters apply in this type of plant, but must be considered also from special angles determined by the process involved.

Apart from the needs of the ware the main combustion requirements are governed by considerations of heat transfer. Until combustion can be maintained in the setting or in muffle flues, heat transfer from the fire gases takes place mainly by convection. Combustion of the coal should be completed in the firemouths with an excess of air. This applies to stages of the firing below about  $700^{\circ}\text{C}$ ., since flame combustion cannot be completed in the kiln.

At higher kiln temperatures heat transfer is mainly by radiation, heat transfer from long flames having an important bearing on the uniformity of temperature distribution. During the higher temperature stages of the firing therefore it is desirable that most of the volatile matter from the coal should be burned among the ware so long as this can be done without detriment to the operation. In muffle kilns this combustion takes place in the kiln flues. The use of a deep fuel bed to provide carbon monoxide is also advantageous during this stage.

For each stage of the firing air control is very important and doors, covers or controllable secondary air flues are provided for this purpose. In exposed positions the provision of glut doors is an advantage to reduce wind effects and to allow the admission of primary air to be controlled.

Combustion conditions that require to be watched in order to attain economical use of fuel are as follows :—

(a) *Water Smoking Period.* Unless the goods have been pre-dried before setting in the kiln, very slow rates of combustion are required during this period because the maximum safe rate of drying may be very low. Slow burning is obtained by maintaining a shallow fuel bed, which remains at a comparatively low temperature because the kiln draught is low and only a small proportion of the air passes through the fuel. The drying is carried out by heated air and some 10–15 times the air necessary for combustion may have to be drawn through the kiln in the earliest stages. This may be decreased as drying proceeds till only about three times the volume of air needed for combustion is admitted when the temperature of the goods reaches  $100^{\circ}$ – $150^{\circ}\text{C}$ . The most satisfactory conditions are obtained when the difference between the temperature of the waste gases and their dew-point is about  $6^{\circ}\text{C}$ .

Artificially induced draught may be employed in the early stages to provide the required excess of air. The potter avoids conditions which might lead to condensation of moisture in the cooler parts of the setting whatever the type of oven, and this is controlled by the way in which the fires are gradually built up.

(b) *Rising Temperature Period.* In this period chemical changes in the ware become important. Chemically combined water is removed and if carbonaceous material is present it must be burnt out. Iron compounds must be fully oxidised. A specific oxidising atmosphere is thus needed during this stage of the firing and accurate control of secondary air is required.

Rates of firing depend on the time required to complete the oxidation before a temperature of  $850^{\circ}\text{C}$ . is reached in the kiln. The most efficient method of firing is that which avoids fluctuations in the temperature-time curve and undue excess of air. This can be assisted by giving the firer or the burner a heating schedule. If a recording pyrometer is used, it is recommended that the ideal heating curve be drawn on the chart as a guide. Fluctuations may be unavoidable with intermittent firing and are not so serious under these conditions provided that black smoke does not issue from the oven over long periods.

In the early stages fires should be built up gradually by small and frequent additions of fuel. Secondary air should be controlled to give approximately 8–12 per cent.  $\text{CO}_2$  in the waste gases. The carbonaceous matter in the ware, which is an additional source of heat, requires air for its combustion. The minimum content of excess air, subject as stated before, to the limiting conditions of the process, gives in this as in other heating processes the maximum economy of fuel. The advantage of preheating the secondary air is all the more marked when an oxidising condition is required to be maintained in the kiln or oven.

At a suitable stage in the firing, when the temperature has reached a point at which combustion of the volatile matter can be ensured by the temperature of the ware, and where secondary air flues are not provided, another method of firing may be applied. This consists in feeding alternate fires, whereby hot secondary air from the intermediate fires can burn the volatiles from the freshly charged fires. It is most important to observe correct timing of the air controls in relation to the time of firing, and also to set the air controls correctly. The appearance of the smoke haze from the chimney stack is a useful indication of combustion conditions where more refined methods are not available.

Only a small proportion of pottery ovens work with an external stack, but observation of the absence of smoke at the outlet of the hovel or cone is possible if the ovens are not enclosed in a building.

(c) *High Temperature Period.* Whilst generally high rates of heating may be employed at this stage and may be consistent with economy, again the technical features of the process determine the practice applied, and a soaking period may be necessary to ensure uniformity of temperature and the completion of the necessary chemical and physical changes in the ware, e.g. vitrification. With heavy firing, if oxidising conditions are required, local reduction may result, and ware be spoilt as well as fuel wasted. This is due to the difficulty of supplying sufficient air to burn the large volume of volatile products initially evolved. Small and frequent firing provides better conditions for the control of air.

In the normal method of firing, the glowing fuel is pushed down into the firemouth and the fresh charge of coal placed at the front and top of the fuel bed. With certain types of firing a steadier rate of evolution of volatile products may be obtained by using the coking method of firing. With this method the fresh charge is placed on a sloping refractory table above the main fire. When

the next charge is due the partially coked fuel is pushed from the table on to the fuel bed proper and the green charge again fed on to the table.

In pottery practice the rate of heating the ware is slowest at the top temperature although the rate of burning the fuel may be highest. Soaking is found to be necessary for most ceramic products in the first fire (biscuit) and is usual in the second or glost fire, but in the decorating kiln there is little or no soaking (i.e. deliberate holding at a constant temperature). In biscuit firing of pottery, fuel additions in the final period are usually heavy and it is generally necessary to maintain oxidising conditions if ware is not to be spoiled and fuel wasted.

Considering the oven or kiln as an isolated unit, i.e. in the absence of some form of recuperator, a high percentage loss of heat in the waste gases is the price which must inevitably be paid for the attainment of high temperatures in the setting, and the price is higher the larger the excess or deficit of air. Thus if the waste gases at the point of exit from the kiln are at a temperature of  $1,000^{\circ}\text{C.}$ , they will carry away 43 per cent. of the heat in the fuel (assumed to be coal with a calorific value of 14,000 B.Th.U. per lb.) even if combustion is perfect, i.e. no excess air. With 50 per cent. excess air they will carry away 60 per cent. of the heat in the fuel.

The important influence of secondary air control can be illustrated in the following manner. If the combustion temperature of the fuel is in the neighbourhood of  $1,400^{\circ}\text{C.}$  and the finishing temperatures are in the range  $1,000^{\circ}$ – $1,300^{\circ}\text{C.}$ , since the useful transfer of heat only occurs as long as the temperature of the heating medium is above that of the goods, a stage must be reached when the hot gases are imparting only a small proportion of their heat to the charge and the chimney loss must be necessarily high. If excess air is increased at this stage a serious loss of heat from the contents of the kiln or oven must occur, since air picks up sensible heat from the already hot mass, which generally has a large surface exposed to the gases.

If on the other hand the excess air is reduced beyond the safe limit consistent with good temperature distribution, a reducing atmosphere causing unnecessary damage to the contents of the kiln may result. Spoilage of material and waste of fuel are thus largely governed by the manner of the air admission to the firing points and the general control of the kiln draught.

Summarising the requirements of fuel efficiency, as in other types of heating appliance, importance must be attached to correct selection and application of the fuel, combustion efficiency, draught control, the maintenance of correct temperature schedules and the uniformity of temperature distribution.

## (2) CONTINUOUS KILNS

Since the drying and preheating is performed by hot air from the cooling zone and hot kiln gases from the firing zone in continuous kilns, main considerations of the firing concern the conditions required in the firing zone. Thus once the required drying and preheating conditions have been determined, these stages of the heating are mainly controlled by fixing the lengths of the zones or parts of the circuit concerned. The drying and thermal requirements of the ware must be met as in intermittent firing, but because more control is usually obtainable much faster firing schedules are usually permissible.

Although most of the heat is recovered from the kiln gases and excess air does not involve the high losses found in intermittent practice, heat losses from these sources are not negligible and warrant some attention. Waste gases should not pass to the stack at temperatures above  $100^{\circ}$ – $150^{\circ}\text{C.}$  unless high draught is required and fans are not available. When fan draught is used waste gas temperatures need not exceed  $60^{\circ}$ – $70^{\circ}\text{C.}$  Heat losses due to excess air involve losses proportional to the waste gas temperature and the excess air

present. The influence of excess air is most marked in the firing zone. Heat distribution in the whole circuit is influenced by the firing conditions used and by the rate of fire or car travel, and conditions in forward zones may have to be altered when these factors are varied. These alterations are mainly effected by adjusting the draught balance.

The firing proper must be considered in relation to the type of kiln employed. Car tunnel kilns in particular are amenable to continuous mechanical firing of solid fuel, gas or oil or electric heating may be used. Firing conditions are easily controlled. Again, since the tunnel is of small cross section, uniformity of temperature distribution is fairly quickly established. The main requirements are: correct draught balance; avoidance of undue excess air, particularly in the firing zone; and adjustment of the rate of car travel, of setting, and of flame or firing conditions, to provide uniformly fired products. Typical continuous kilns are shown in Fig. 153.

With top fired Hoffmann and transverse arch kilns most efficient firing conditions are obtained when small dry coal is fed continuously so that much of the combustion may be completed before the coal reaches the kiln floor. Thus, when mechanical stokers are not used, there should be very frequent additions of fuel at not more than 15-minute intervals. Air control is again important and draught requires special attention. Other factors which influence the firing concern the type of setting employed, the number of cooling chambers and the number of forward chambers in the circuit.

The special requirements of grate and fender fired kilns, such as the Belgian and transverse arch chamber types, mainly concern control over combustion rates obtainable. The draught in the firing zone must be adequate to draw the maximum required amounts of primary air through the fuel bed without involving undue excess of secondary air in the chambers. For this reason the fuel bed must have a low resistance to air flow. Graded coals are therefore used and since the grates cannot be cleaned during the firing the coals should have low ash content. Generally coals of  $\frac{1}{4}$ –1 inch size are suitable for fender firing and coals of 2–3 inches size for the firing of Belgian kilns. Good forward heat should be aimed at so that the minimum amount of coal is needed to reach the finishing temperature. Frequent firing at 15–20-minute intervals is again desirable, and a thin level fuel bed should be maintained. Long flame coals are specially desirable because the whole of the chamber must be uniformly heated by coal fed to each grate, the distance between grates being 11–15 feet.

Different methods of firing car tunnel and annular tunnel kilns are used in the pottery industry. Contamination due to the effect of atmosphere is overcome either by muffle heating or firing in saggars, by the choice of suitable fuel or by electrical heating. Counterflow types of continuous kilns are also employed. With open-flame heating, producer gas, town gas or oil firing provides conditions which are readily controlled, and once the air requirements have been properly adjusted to give satisfactory combustion conditions only minor adjustments are required. With these fuels multiple burners may be used to increase the uniformity of heat distribution. Although costs are high, electrical heating offers advantages, and in particular thermostatic control over finishing conditions is readily applied. For muffle firing the air admitted to the firing zone need not much exceed theoretical requirements.

## SETTING

The arrangement of the ware in intermittent kilns, in chambers of continuous kilns, or on the bogies or moving hearth of car tunnel or annular kilns, is equivalent in its effect on fuel consumption to the loading of other types of heating appliances. In kilns and ovens the material heated has thermal properties of the same order as the brickwork used in the construction of the

heating chamber. This means that rates of heating are necessarily low and accordingly the character of the setting is critical in permitting a suitable circulation of hot gases, and at the same time giving an optimum rate of heat penetration in keeping with the technical requirements of the process worked.

The hot products of combustion are capable of a rate of heat release to the contents of the heating chamber which is very much higher than the capacity of the material heated to absorb it. The character of the setting is therefore an important factor in the determination of efficiency. Specification of the optimum setting density for all materials fired in the different kilns is not possible here because of the variable requirements. In general, however, ware which is not pre-dried before setting in the kiln, or ware containing much carbonaceous matter must be set more openly than other green products.

In pottery firing a special problem concerns the relative efficiency of placing in saggars or firing in muffles, where contamination with sulphur gases must be precluded. Much empirical work has been carried out on problems such as these, on methods of setting, and on setting density, but further research is required to determine optimum conditions. The question is important because fuel consumption, throughput, uniformity of treatment and of heat distribution, may be greatly influenced by the setting.

## FACTORS AFFECTING FUEL EFFICIENCY

The main general factors affecting fuel efficiency concern :—

- (a) Control of air to meet combustion requirements and the special requirements of the ware.
- (b) The suitability of the fuel or adaptation of firing methods to suit the fuel which is available.
- (c) The maintenance of the shortest safe firing schedules and obtaining maximum throughput from the plant. In this connection as much of the production as possible should be fired in continuous kilns when they are available.
- (d) The use of optimum setting density for the goods fired.
- (e) Control over the firing. This includes methods of baiting, temperature and draught control.
- (f) The use of insulation and attention to structural features.

The dimensions of firemouths, grates, fire arches, gluts and other pertinent structural features which may influence the flow of hot gases all fall into the class of controllable factors, which at rebuilds can be adjusted to improve performance. They are all dependent upon the process operated and local conditions ruling. Again no hard and fast rules of a general type can be laid down beyond an elaboration in detail of the principles which it has been the aim of this work to outline, in particular in the chapters on combustion, heat transmission, the flow of gases, and refractory materials. For details of structural technique reference should be made to specialist volumes and technical journals.

In heat insulation in kilns and ovens noteworthy contributions have been made to the more efficient use of fuel by the use of heat insulating materials both as external layers and in the internal structure. Thus the use of hot-face insulating material in intermittent kilns as an inner lining and for crowns, together with insulation beneath the kiln floor, has resulted in a considerable reduction of loss of heat in the structure, in producing greater uniformity of heating, and in reduced duration of firing for those wares which may be fired without harm at increased rates. A limitation arises in the practicability of using hot-face insulating materials for the highest temperatures. Saving in

fuel amounting to 10 per cent. and upwards of the original fuel consumption have been claimed as a result of using such materials.

The improvement of the structural features of large kilns and ovens, either of the tunnel or other unique design is a problem of a special type and its detailed discussion is a matter for a more specialist work.

Attention is directed to the value of measurements of temperature, draught and gas analyses, and to records of all factors concerned with firing. Measurements and records frequently enable fuel savings to be made because they emphasise the importance of economy to the firemen and provide objectives to be attained.

### FURNACES AND FURNACE CONTROL IN THE GLASS INDUSTRY

In the manufacture of glass various types of furnaces and heating appliances are employed, designed each for its particular purpose, but by far the greatest proportion of the fuel used is that for heating the glass melting furnaces. Of these there are two general types known, respectively, as tank furnaces and pot furnaces. A relatively small amount of heat is used for the annealing of the glass or glassware.

#### MELTING FURNACES

*Tank Furnaces.* Tank furnaces consist essentially of a tank or trough, the bottom and lower sides of which are constructed of some suitable refractory material in the form of large blocks. Rising from the walls of the tank proper are the upper walls and covering the tank is an arch. The whole of this superstructure is usually made in silica brick.

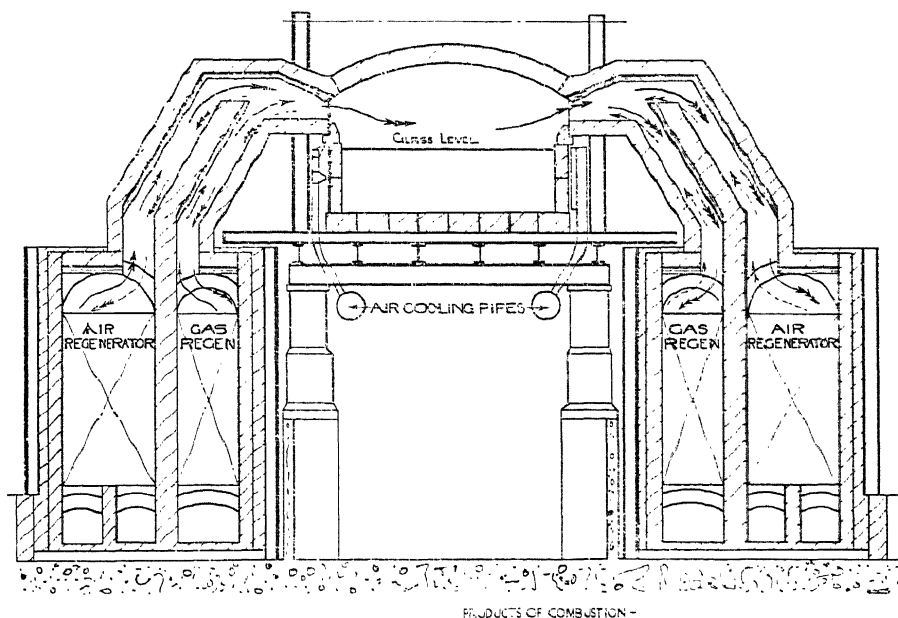
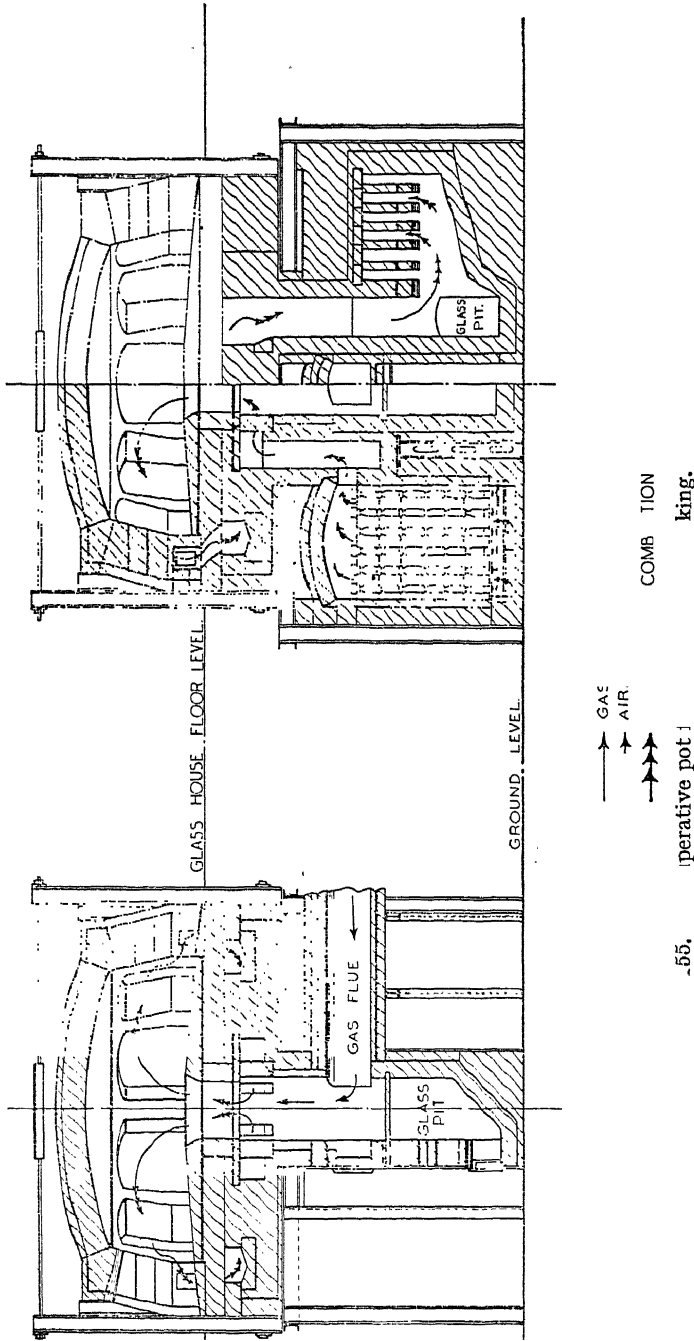


FIG. 154. Cross-fired regenerative glass melting tank, fired with producer gas.



A burner or burners are provided to supply flame for melting the glass, the flames being either projected down the length of the furnace ("end-firing") or across the width ("cross-flame firing") and flues are provided for the waste gases. These burners, including the means of supply of the requisite air, vary in design according to types of fuel and other circumstances. In all but very small furnaces, as much as possible of the heat in the waste furnace gases is recovered in recuperator or regenerator chambers, such recovered heat being employed to preheat the air needed for combustion; and, in furnaces fired by producer gas, two regenerator chambers are usually employed, one for preheating the air, the other for the gas (cf. pp. 153 and 385).

Tank furnaces include day tanks and continuous tanks. Day tanks may be quite small, in which case recuperators or regenerators may be omitted, or may hold up to, or even more than 20 tons of glass. They usually contain no bridge and their operation resembles that of pot furnaces in that periods of melting and of working alternate.

The modern continuous tank furnace is almost invariably of the regenerative or recuperative type, recuperators being employed more generally for smaller furnaces. A typical cross-flame regenerative bottle glass tank is shown in Fig. 154. The tank is divided into two compartments by a double-walled bridge, the larger of the two compartments being termed the melting chamber and the smaller the working chamber. The depth of the glass is maintained constant by charging in raw materials at the melting end equivalent to the rate at which glass is withdrawn from the working end.

In tank furnaces for the continuous production of ordinary drawn sheet and rolled plate glass there may be three or even four compartments divided in most furnaces by floating bridges, and cross-flame firing is generally employed.

Auxiliary heating is applied to the discharge or working end of most tanks for controlling the glass temperature at the machines.

The average glass bottle tanks used in Great Britain range from 30 to 60 feet in length, 2 feet 6 inches to 3 feet 6 inches in depth according to the type of glass, and have a dead weight capacity between 100 and 400 tons of molten glass. Sheet and plate glass tanks are substantially larger and may contain 800 to 1,000 tons of glass.

*Pot Furnaces.* This is the most ancient type of furnace, in which the glass is melted in fireclay vessels. The pots themselves are usually of fireclay, or of fireclay and sillimanite, may be open or provided with a hood, and may vary in size from a few pounds up to 2 tons capacity. A so-called pot furnace may contain from one to twenty-four such fireclay pots. There are still a few pot furnaces employed which are either direct-fired or only semi-gas-fired.

In circular furnaces the flame, from whatever source it is derived, rises through a circular aperture or "eye" in the hearth or floor on which the pots stand in a ring, and is then drawn downwards from the dome of the furnace through flues constructed in pillars between the successive pairs of pots or crucibles. There are, however, other types of firing, and other shapes of furnaces than circular. Plate glass furnaces, mainly, if not exclusively, of the regenerative type, are usually rectangular in shape, have a row of pots down each side, and end ports give a sweep of flame down the length. In England the newer pot furnaces employed in the domestic glass industry are mainly recuperative and a typical furnace of that type is shown in Fig. 155.

## ANNEALING FURNACES

Annealing furnaces or lehrs, as they are called in the glass industry, are comparatively long and narrow structures of tunnel formation through which the glass or glassware passes more or less continuously. Heating facilities are provided in all lehrs and are applied either directly or indirectly. The source



of heat may be any fuel—gaseous, liquid, solid or electricity. Many lehrs are built of brick with metal conveyors of various designs. The most modern bottle and press ware lehrs are constructed entirely or mostly of metal, are well insulated and employ the mattress type of conveyor. Some lehrs require no fuel or energy for a wide range of operating conditions, relying entirely on the heat in the glassware as it comes still very hot from the machine.

## GENERAL METHODS OF PROCEDURE IN GLASS MELTING :—

### TANK FURNACES

(1) *Charging the Furnace.* The mixed raw materials, comprising sand, limestone and soda ash and sometimes including a proportion of scrap glass or “cullet,” known as “batch,” may be charged into the furnace by :—

- (a) Hand shovelling.
- (b) Semi-automatic means.
- (c) Fully automatic means.

In (a) and (b) the batch is introduced at frequent intervals in amounts consistent with the rate of withdrawal of glass from the furnace. In (c) the feed is similarly arranged, but is steady and continuous. The difference between methods (b) and (c) is that the automatic mechanism enables charges to be made at regular intervals of time. Completely automatic methods enable the furnace to be kept closed, thereby guarding against loss of heat and variation in temperature, and thus making for greater fuel efficiency.

Glass batch is “dusty” and avoidance of choking of regenerators is a matter for special consideration in the design of glass tank furnaces. Slag pockets, etc., are introduced and the modern tendency leans to the use of “briquetted” batch in order to curtail batch carry-over.

(2) *The Melting and Refining Processes.* Glass melting in tanks may be divided into three stages. First, the batch is pushed into the furnace and melted into a frothy bubbly glass. Second, this glass is refined, which means holding it at a temperature sufficiently high to get rid of the gas bubbles (partly by solution and partly by bringing them to the surface), and also to bring about the interdiffusion of the molten materials to make the resulting glass clear and homogeneous. Finally, the glass flows to where it is gathered by the glass makers on the machines and in so doing it is cooled to a temperature suitable for the particular process in use.

The first two stages just described are carried out in the melting chamber of the furnace and, as refining takes the longer time and requires the highest temperature, the major part of the space must be reserved for it. This means that in a glass tank all batch piles, other unmelted batch and scum must be confined to the smallest possible space at the batch entry end of the furnace. From 60–80 per cent. of the glass surface in the melting end should be left free or refining and this part of the furnace should be the hottest.

The best test of whether or not a furnace is producing good glass is the appearance of the glass as it is drawn from the working end. Consequently, providing there are no local conditions in the working end, feeders, forehearth, etc., which may be causing seedy glass, and providing other obvious precautions are observed, the appearance of seedy glass is an indication that the temperature is too low or that glass is being withdrawn from the working end too rapidly. The correct temperature is the lowest which will give glass of the required quality and in an amount sufficient for a reasonable production schedule.

(3) *Conversion of the Glass into Glassware.* The glass is supplied from the working end to one of several operating units, which may comprise workmen or

may be machines. According to circumstances, the operating units may consist entirely of workmen who carry out the shaping of the glass, or a combination of workmen and machines, or machines only. The machines in turn may all be of similar types and capacity or may vary. Although in practice such a distribution may be impracticable, it is an ideal to aim at to discharge the glass at rates as nearly the same as possible from all points round the working end where operation is proceeding.

Continuous tank furnaces are in operation throughout the twenty-four hours of the day and, in most works, for seven days in the week; and they remain in operation for periods of ten months to two years, or even longer; or, in general, until major repairs must be carried out.

## POT FURNACES

(1) *Preheating the Pots.* The pots, which may have a bottom thickness from 3-5 inches and a corresponding wall thickness, are carefully preheated in an auxiliary furnace, known as a pot arch, over a period varying from five to fourteen days depending on the size of pot, the precision of the control exercised and the temperature to which they are fired before transfer, at their maximum temperature, to the melting furnace. Some modern pot arches are constructed with recuperators and can be raised to 1,300° or 1,400° C. In the simpler types the maximum temperature employed is 1,000°-1,100° C.

(2) *Charging the Furnace.* Each individual pot has to be charged. This is usually done by hand, but in plate glass pots from a charging machine attached to a truck moving on rails parallel to the furnace length. For most pot furnace work waggons holding some 8-10 cwt. of batch and cullet are employed and the pots are charged from shovels.

In furnaces in which melting by night and working by day is the system followed, the pots should all be emptied by a fixed time in the afternoon and after being brought up to melting temperature again, all will be charged to the same extent. With furnaces in which melting and working are proceeding side by side during the twenty-four hours, the quantity of material charged will depend on the working schedule. For example, some pots may be worked on small articles with the result that they are only half emptied over the normal working period, and whilst from one viewpoint it is desirable to work as much as possible, the prolongation of the work from one pot may interfere with the successful working of pots in the immediate vicinity which are to be worked on the next shift. A pot will then be filled once only, as this quantity of charge will fill the pot and make it ready for work at a time when adjacent pots are founding. When pots are worked low, two and sometimes three charges of material are necessary before they are full again. The period which elapses between the separate charges depends upon the size of pot, size of charge, type of glass, etc. Usually eight to ten hours is necessary between the individual charges.

(3) *Melting and Refining Process.* The pots may be either of the open type or the closed (that is, covered or hooded) type. Open pots are employed for melting plate glass and sometimes for electric light bulbs. For most other purposes in Great Britain covered pots are employed. The glass in open pots is heated both through the walls of the pot and by flame which passes over its surface. With a covered pot, the heat is supplied through the walls and by radiation from the crown. Melting is slower in closed than in open pots.

There are three stages in the complete preparation of the glass for working. In the first stage the materials must be completely melted; in the second, time and usually higher temperature are needed to refine the glass; whilst in the third stage the already refined glass is slowly cooled to an appropriate temperature level for gathering. The aims are similar to those in tank furnace